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STEAM LOCOMOTIVE DESIGN.

By D. PATRICK.

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FOREWORD.

IN this paper the design of component parts of steam locomotives is discussed, embracing all the more important details, with the exception of the boiler and its attachments. The preliminary design of a locomotive as a whole has already been well treated in the paper by Mr. T. Grime, published by the A.E.S.D. some years ago, and the present paper begins where he finished—at the stage when the wheel arrangement, size of cylinders, etc., have been settled.

In the space available the author has confined his attention to modern locomotives of normal design. In the section on valve gears the Walsehaert's type only has been dealt with, since such types as Stephenson's link motion or Joy's radial gear are seldom applied to new designs.

The detailed design of locomotives is treated in various books of reference, some of which are rather out-of-date. Much of the following matter, therefore, must be familiar to many people. The author has endeavoured to treat the subject in such a way as to make it as directly useful as possible to the practical draughtsman, and he hopes his efforts may meet with his approval.

Cylinders.

In modern practice two-cylinder engines are usually designed with the cylinders outside the frames. The earlier practice of placing them between the frames, while admittedly giving a somewhat steadier running engine, has the disadvantage of requiring an expensive crank axle, and also sets a definite limit to the size of cylinders and valves which may be accommodated in the width available. With outside cylinders there is still a limiting size, due to the loading gauge, and three or four cylinders are frequently employed to get the desired power. (Also, two very large cylinders produce a less even turning effort and generate greater unbalanced forces than three or four smaller cylinders).

Considering the most usual type—an outside cylinder for a two-cylinder locomotive—the best modern design embodies steam-distribution by piston valves placed above the cylinder barrel and operated by Walschaert's valve gear, with the piston valve arranged for inside admission.

The vertical distance from the cylinder centre to the piston valve centre should be as small as possible, in order to keep down the length of the steam ports. This dimension is usually fixed by the space required for the cover flanges on the cylinder and steam chest respectively. On the end view, the piston valve centre should be some inches further out than the cylinder centre if possible, this dimension being fixed from the motion arrangement.

The ideal design is, in a plan view, to have the piston valve and all the motion rods right back to the eccentric crank on a common centre line. The loading gauge of the railway sometimes prohibits this, in which case the valve must be moved in, and off-sets put in the valve gear as required. The horizontal distance from the frame to the centre of the cylinder is also derived from the motion arrangement, representing the space required for wheels, coupling rods, connecting rod, and crosshead. These are all discussed later under motion arrangement.

Piston valves may be of "inside" or "outside" admission types. The "inside" admission type, in which the steam is admitted to the ports from the centre of the valve chest and exhausted at each end, is now almost universal, having the advantage that the valve spindle glands and steam chest covers are only subjected to exhaust steam pressure. Consequently no elaborate packing is necessary on the valve spindle. The steam ports should run straight up into the steam chest, thereby giving an easy path for the steam and keeping down the clearance volume to a minimum. This necessitates a long steam chest to accommodate the exhaust passages, which project beyond the cylinder wall at each end and are carried round and united conveniently at the back.

Thickness of Metal in Cylinders.

According to Pettigrew,

$$\text{Thickness of cylinder barrel, } t = \frac{P \times D}{5000} + 0{\cdot}6''$$

where t = thickness of barrel in inches.
P = boiler pressure in lbs. per square inch.
D = diameter of cylinder in inches.

This formula includes an allowance for re-boring, and gives a thickness slightly greater than cylinders are now usually made, especially if weight saving is of importance.

$$t = \frac{P \times D}{5000} + 0{\cdot}4'' \text{ is often considered adequate.}$$

The thickness of the main end walls is generally 1″ to $1\frac{1}{4}$″ according to size of cylinder, and the walls of the steam chest ports and passages $\frac{3}{4}$″ to $\frac{7}{8}$″, according to size and pressure. The walls of exhaust passages are generally of the same thickness as the steam

passages, for, although they do not withstand live steam pressure, it is often convenient, for practical reasons, to submit the whole of the cylinder to the hydraulic test.

To withstand the rapid alternation of piston thrust it is most important that cylinders should be securely bolted to frames with driven bolts. In some cases a spigot is provided on the flange, fitting tightly into a hole in the frame, thus relieving the stress on the bolts. For large cylinders this feature is desirable. It is essential that the spigot should be a good fit in the frame. Where bolts only are used, their total cross-sectional area works out at approximately 1·2 to 1·8 square inches per ton of piston load. The diameter of the bolts varies from 1″ to $1\frac{1}{2}$″.

If d = diameter of bolt,
thickness of flange to frame = $d + \frac{1}{8}$″ (minimum).
and distance of bolt from edge of flange = $d + \frac{1}{4}$″ (minimum).

To quote an actual example, of conventional design, as represented by Fig. 1 :—

Cylinder $15\frac{1}{2}$″ diam. × 20″ stroke. Piston valve 8″ diam.
Working pressure, 180 lbs. per square inch.

Thickness of	barrel	= $1\frac{1}{8}$″
,,	,, end walls	= 1″
,,	,, steam chest, steam ports, and exhaust passages	= $\frac{3}{4}$″
,,	,, flange to frame	= $1\frac{1}{4}$″
Diameter of	bolts to frame	= $1\frac{1}{8}$″
Number of	bolts to frame	= 22
Distance of bolt from edge of flange		= $1\frac{3}{8}$″

To fix the length of the cylinder barrel, the width and type of piston must be determined. Pistons usually follow standard practice on the railway concerned. The width of a piston head varies from about $2\frac{1}{2}$″ to $4\frac{3}{4}$″, and two or sometimes three cast iron rings are fitted. Some railways use rings $\frac{1}{2}$″ to $\frac{7}{8}$″ wide, with or without a groove on the face, while others use rings $\frac{1}{4}$″ or $\frac{5}{16}$″ wide and two only. (See Fig. 1). The two narrow rings appear to be now the most favoured.

Common types of piston head are shown in Fig. 2. They are generally of cast iron, or cast steel with a cast-iron rim. Pistons made entirely of cast steel are not much favoured, as they may cause excessive wear or scoring of the cylinder barrel.

The cylinder barrel should be so arranged that the rings sweep the whole of the parallel bore, to avoid forming a ridge. The position of steam ports in the barrel also should be such that the body of the piston does not obstruct them unduly when at the ends of its stroke.

The cylinder covers follow the contour of the piston, with a clearance of $\frac{1}{4}$″ to $\frac{3}{8}$″. Where the standard of maintenance is high $\frac{1}{4}$″ may be sufficient. In abnormal cases where the cylinders are

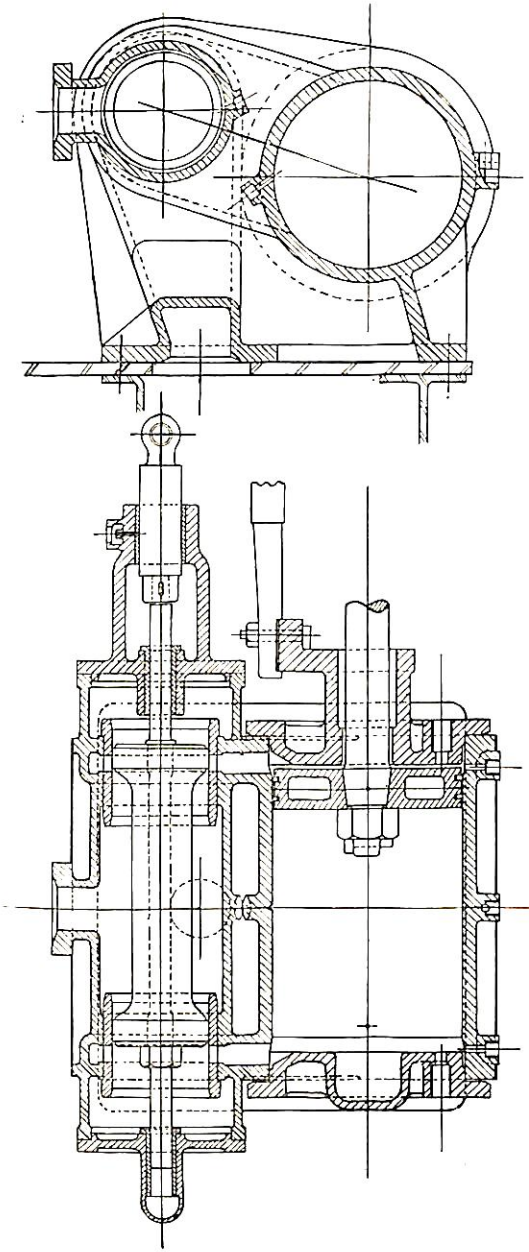

Fig. 1.—Cylinder Arrangement.

steeply inclined (as on some inside cylinders), the clearance is sometimes made $\frac{7}{16}$" to allow for variation in piston travel caused by rise of the axle.

The area of steam ports is important. If they are made large to get a low steam speed, the clearance volume becomes excessive, so a compromise is made. According to modern practice the net area of each steam port is about one-ninth to one-tenth of the cylinder area, this being sufficient to pass the exhaust. It is slightly less than the bore of the exhaust pipe. Pettigrew quotes an area as low as (·07 × cylinder area), but this is small compared with recent practice. With straight ports of the area suggested, the clearance volume (*i.e.*, the space between the piston and cover, plus the steam port right up to the working face of the piston valve liner) does not exceed about 8% of the swept volume of the cylinder, and may sometimes be very much less.

The length of the port may equal (·8 × cylinder diameter), and at least one bar should be cast in the port and bored with the cylinder to prevent piston rings being trapped in ports during erection. From this the required width of port is obtained. Finally, the width of port through the piston valve liner is made $\frac{1}{4}$" less than this, providing a facing on the cylinder which is machined to line up with the ports in the liner, which are also machined. The net area through the ports in the liner should be somewhat greater than the port area already determined.

A main steam pipe of area $= \frac{\text{Cylinder Area}}{17 \text{ to } 18}$ represents average practice. On the basis of a piston speed of 1000 feet per minute, this corresponds to a steam speed in the pipe of 17,000 to 18,000 feet per minute. However, such a high velocity is not attained, as with such a piston speed the cut-off point is early in the stroke, so that admission takes place mainly while the piston is accelerating from rest.

For the bore of the exhaust pipe from the cylinder, the usual area is about one-seventh to one-ninth of the cylinder area—preferably the former. The exhaust passages in the cylinder itself, being of a varying cross-section, roughly rectangular and containing bends, should have a minimum area rather greater than the exhaust pipe to ensure the free passage of exhaust steam.

Cylinder Covers and Piston.

Fig. 2 shows some common types. Their design is largely empirical, as accurate stresses cannot be arrived at except for the studs. The cover flanges are usually about the same thickness as the cylinder barrel, or slightly less. For the pitch of studs, Pettigrew gives—

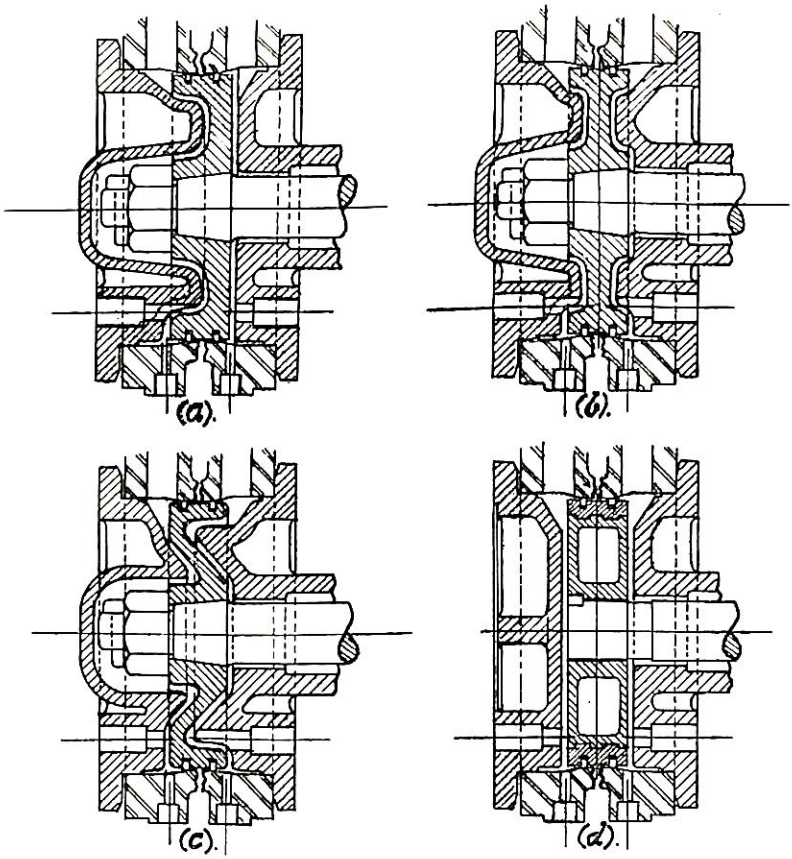

Fig. 2.

$$\text{pitch} = \sqrt{\frac{t \times 100}{p}}$$ where t = thickness of flange in sixteenths of an inch.
p = boiler pressure in lbs. per sq. inch.

An inspection of actual examples, however, shows that the pitch is frequently about $\frac{3}{8}''$ to $\frac{1}{2}''$ more than that given by this formula. For the diameter of the studs, a stress of about 5000 to 7000 lbs. per sq. inch is allowed, based on steam pressure acting on a diameter touching inside of studs.

The back cylinder cover makes a faced joint on the cylinder, as it carries the slide-bar and packing gland, and for that reason it is usually about 1″ thick in the body, with ribs on covers of large diameter. The joint at the front cover is commonly made with a

seamless copper ring, and for this cover the thickness of metal may be $\frac{5}{8}$" or $\frac{3}{4}$", with ribs to the number of about one per 4" of cylinder diameter.

For piston heads, the least thickness is given by Pettigrew as $\frac{\sqrt{D}}{3}$ where D = diameter of cylinder in inches for types (*a*) and (*b*), Fig. 2. For type (*c*) ·8 of this, and for box pistons (typd *d*) ·5 of this for each side.

All these figures refer to cast-iron pistons and covers. Where cast steel is used the values found may be multiplied by ·8 in each case.

Although cast steel pistons are not favoured, it is claimed that mild steel or alloy steels are satisfactory, and a large reduction in reciprocating weight is possible with their use, as the body thickness need only be about half of the corresponding thickness for cast iron. Conical pistons give the maximum strength with the minimum weight. On the other hand, a box piston gives the minimum clearance volume, especially the type used in the Indian standard designs, where the piston head is screwed on the piston rod, giving a flush finish.

The foregoing data for pistons and covers serve as a guide, but practice varies considerably and reference to existing good designs is essential.

It is usually advocated that cylinder covers should not be made excessively strong in proportion to the cylinder, as in the event of a breakage due to priming or some other cause, it is easier to replace a cover than a cylinder body.

Piston Valves.

The design of piston valves and valve travel will next be considered. Types of piston valves are many, and they vary greatly in design. Some consist of two simple heads fixed on the valve spindle, each carrying the necessary cast-iron rings. In others, the heads are united in a hollow casting, or alternatively steel heads are welded to a steel tube, the latter form being much lighter. This hollow type has the advantage of allowing the steam from the cylinder to exhaust partly through the centre of the piston valve and so to utilise both exhaust passages. Against this may be put the fact that the exhaust steam in the hollow valve must have some cooling effect on the live steam in the steam chest. Some types have pressure release valves incorporated in them. There are many variations in the size and mounting of the rings. Reference to existing practice is essential, and illustrations of various types will be found in "The Locomotive of To-day" (a collection of reprints issued by the Locomotive Publishing Coy.), as well as in other similar books.

The principal point—the diameter of the valve for a given cylinder—is not even standardised. The following sizes are suggested by Selby (Proc. Inst. of Loco. Engineers) :—

Cylinder Diameter,	12″ to 14″	14″ to 16″	16″ to 18″	18″ to 20″	20″ to 22″	22″ to 24″
Piston Valve Diam.,	7″	8″	9″	10″	11″	12″

The width of port in the piston valve liner being already fixed, the maximum opening to live steam will be about $\frac{1}{4}$″ less than this.

The "lead" (*i.e.*, the amount by which the valve is open to steam when the crank is on dead centre), is usually $\frac{1}{8}$″ or $\frac{3}{16}$″.

Determination of Valve Travel.

From the preliminary design work the maximum cut-off, in percentage of stroke, will be known. With this information (opening to steam, lead and maximum cut-off), the necessary valve travel may be obtained by trial from a simple valve diagram. (See Fig. 3). The steam lap of the valve (the amount by which the steam edge overlaps the port when the valve is in its mid position) and the valve travel, are the two adjustable factors. They must be mutually adjusted by trial until the desired lead, opening and cut-off are obtained.

In general, a long valve travel gives the best results, as the valve has a higher linear velocity and therefore opens and closes the ports more quickly. Thereby, it reduces the wire-drawing effect as the valve is just opening or closing. By a long travel is meant about 5″ to 6″, in contrast to a short travel of about 4″. Large American engines sometimes have about 8″ valve travel. This trouble of wire-drawing, with consequent loss in effective steam pressure, is one point on which poppet valves have an advantage over piston or slide valves.

Considering now the exhaust side, if the valve has no exhaust lap, as is commonly the case, the points of release and compression fix themselves as shown on Fig. 3. Sometimes the valve is given a small "negative lap," *i.e.* both ports are open to exhaust by this amount when the valve is in its mid-position. This reduces the compression period, and also causes the release to occur earlier, as shown by the dotted line on Fig. 3. This feature is usually found on high-speed locomotives, where it is desirable to keep the compression period within reasonable limits when working with an early cut-off.

When the locomotive is under way, the cut-off is reduced by shortening the valve travel, and since the lap and lead are fixed, this makes the release and compression occur earlier also, which is unavoidable. To control the cut-off in a cylinder independently of

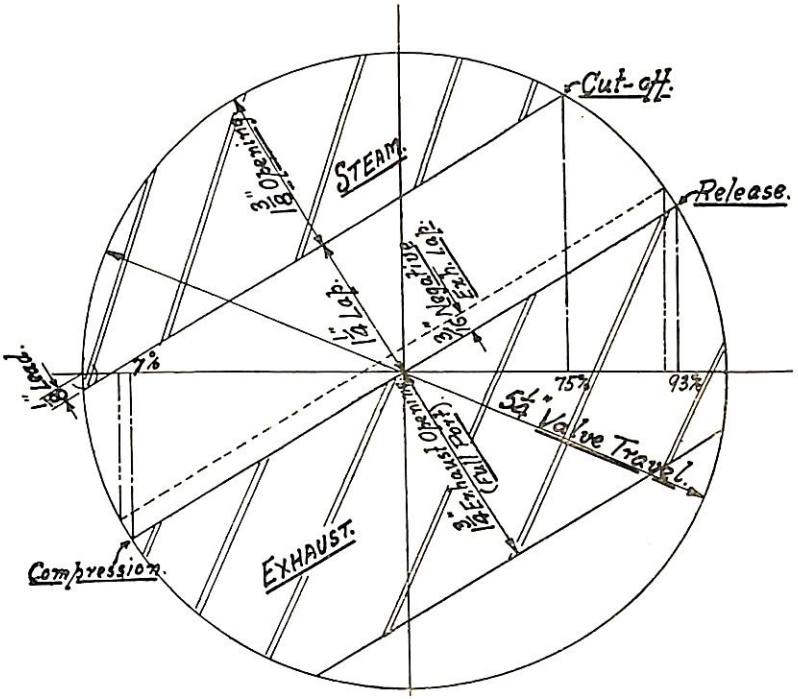

Fig. 3.—Valve Diagram for Full Gear.
Dotted Line shows effect of Exhaust Lap.
(Obliquity of Connecting Rod is neglected.

the exhaust events, one must have recourse to cam-operated valves, such as those of the well-known Lentz or Caprotti types.

Note that in Fig. 3 no corrections are made for the angularity of the connecting rod, due to which, piston positions on the out-stroke differ from those on the in-stroke for corresponding crank positions. This is done deliberately, since the valve diagram is only used at this stage to obtain approximate figures, and is accurate enough as it stands for that purpose. Valve diagrams of all kinds are fully discussed in most text-books on valve gear.

Returning to the cylinder—the steam chest, covers, and valve spindle guide can now be designed with due regard to clearances of the valve and spindle.

The piston valve heads work in renewable cast-iron liners about $\frac{3}{4}$" thick, pressed in. The liners may extend to the end of the steam chest, with large ports for the exhaust, and be located endways by the covers, or short liners just covering the travel of the piston valve head may be used, located by dowels through the steam chest wall. Steam ports in the liners are accurately machined, and are

arranged in a series of triangular or rhomboidal holes, designed to prevent uneven wear on the piston valve rings. The edges of these holes are rounded on the outside to assist steam flow, and the holes in the liners for exhaust (in extended liners) are rounded on both sides.

Attention should be paid to the desirability of making exhaust passages in cylinders as simple as possible, free from abrupt changes of section and from dips where condensate could collect.

A drain pipe from the steam chest is led to a drain cock operated simultaneously with those on the cylinder barrel.

Auxiliary Steam Ports.

Locomotives are sometimes designed to operate with a maximum cut-off of 50% to 65%, and to assist such locomotives when starting, it is a common practice to cut a small auxiliary port through the liner, so placed that steam is admitted through it up to about 85% of the stroke. The port is about 2" long and $\frac{1}{8}$" wide, and is led behind the liner into the main port. It should be on the outside of the steam-chest, with an inspection plug opposite, so that it can be kept clear. In operation, this port functions at starting, but as the speed rises the amount of steam passed becomes negligible.

Inspection plugs opposite the main ports in the liner are useful when setting the valves, though not essential.

Cylinder Lubrication.

The cylinder barrel has an oil feed led to the top centre. Piston valves may have a feed to each head, drilled through the liners, as in current German practice, or one feed only to the steam pipe, made to mix with the steam to form an oily vapour, which also helps to lubricate the cylinder barrel. The latter is more usual on British-built locomotives. The oil supply may be from a hydrostatic sight-feed lubricator in the cab, which has a steam jet and delivers the oil in the form of an emulsion, or from a mechanical lubricator, which passes the oil through a steam-jet atomizer device known as an anti-carboniser for a locomotive using superheated steam.

Other fittings on cylinders include relief valves on each cylinder cover ; bye-pass valves arranged on the steam ports and connected to the steam chest, or on the bottom of the cylinder barrel ; plugs on cylinder covers and steam chest for indicator connections, if required ; and drain cocks from cylinder and steam chest. Such fittings are of standard types for any given railway.

Cylinders fitted with poppet valves will not be discussed here. Numerous drawings of such cylinders and their operating gear have been published in the technical press in recent years.

Working pressures in common use vary from 180 lbs. to 250 lbs. per square inch, and superheating is now almost universal on main line locomotives. On shunting locomotives, superheating is of doubtful value.

Motion Arrangement.

When drawing a motion arrangement, the procedure is as follows.

Starting with the wheels, the width between tyres and their contour are standardised for the railway concerned. An assumption must be made regarding the distance the wheel boss and balance weights project beyond the tyre. This is to be verified later.

The size of the coupling rod bearing on the driving crank-pin is next required. When calculating bearing pressures on coupling rods for a two-cylinder locomotive, it is assumed that each wheel on one side absorbs an equal proportion of the maximum turning effort from one cylinder (piston load × crank radius). Inspection will show that the bearing load on a coupling rod on the *driving crank-pin* is—for a four-coupled locomotive = $\frac{1}{2}$ × piston load.

,, six-coupled locomotive = $\frac{2}{3}$ × ,, ,,

,, eight-coupled locomotive = $\frac{3}{4}$ × ,, ,,

,, ten-coupled locomotive = $\frac{4}{5}$ × ,, ,,

Or, if n = number of coupled *axles*, bearing load on coupling rod at driving crank-pin $= \frac{(n-1)}{n}$ × piston load.

Also, bearing load on each of the other crank-pins

$$= \frac{\text{Piston Load}}{n}$$

A bearing pressure for the driving coupling rod bearing of 1200 to 1500 lbs. per sq. inch of projected area may be allowed, and the bearing made about $\frac{1}{2}$" to $\frac{3}{4}$" bigger in diameter than the connecting rod big-end bearing.

Similarly, for the connecting rod big-end bearing, taking full piston load, a bearing pressure of 1500 to 1900 lbs. per sq. inch is allowed, and the length of the bearing is made equal to its diameter, or slightly less.

Crank Pins.

Next, calculate the bending stress in driving crank-pin, taking a bending moment = (piston load × distance from centre of connecting rod to face of wheel) or if the diameter of the crank-pin in the wheel seat is not greater than that of the coupling rod bearing, the bending moment may be taken to a point about $\frac{1}{2}$" into the wheel seat. A stress up to about $8\frac{1}{2}$ tons per sq. inch with Class D steel is permissible. (Steel to B.S.S. 24/8D, 40-45 tons tensile). This represents the conditions when the crank is on dead centre and the coupling rods are assumed to be taking no load.

The other crank-pins are similarly treated. It is customary to make their diameter such that if the full piston load be assumed

to act at the centre of each pin the bending stress will not exceed that of the driving crank-pin. The diameter necessary for bearing pressure generally ensures this.

In the American Locomotive Company's Pocket Book it is further stipulated that

$P \times V$ = not more than 1,100,000.
where P = bearing load in lbs.
and V = Rubbing velocity in feet per minute

between crank-pin and bush when the locomotive is travelling at a speed in miles per hour, N, = number of inches in diameter of driving wheel.

E.g., If wheel diameter D = 5 feet.
assumed speed N = 60 miles per hour = 5280 feet per minute.

$$\text{Revs. per minute} = \frac{5280}{\pi \times D}$$

If diameter of crank-pin = d feet

Velocity V = revs. per minute $\times \pi d$

$$= \frac{5280}{\pi D} \times \pi d = \frac{5280 \times d}{D} \text{ feet per minute.}$$

The coupling rods should lie on a common centre line throughout, hence the length of all coupling rod bearings must be equal or else suitable bosses must be made on the wheels, other than the driving wheel.

Driving Axles.

The distance between the cylinders being now fixed from the driving crankpin, the diameter of axle can be determined. Complicated bending moment diagrams can be drawn for this purpose, but the following method (also to be found in the American Locomotive Company's Pocket Book) is recommended as being simple and direct. Its results should bear checking by any other method which may be employed. (The following notes refer to straight axles for outside cylinder locomotives).

The principal load on the driving axle of a steam locomotive is that due to the horizontal piston load on the crank pin. The bearing load on the journals is small in comparison, and acts vertically on the axle. Unlike carriage axles, which are subject to a stress due to wind pressure acting on the flat sides of the vehicle, causing a lateral thrust between the wheel flange and the rail, an ordinary locomotive does not present any large flat surface. Further, its centre of gravity is lower than a carriage, so this factor is not a vital one.

Taking a bending moment, M = (piston load × horizontal distance from centre line of cylinder to centre line of journal) and a twisting moment T = (piston load × crank radius),

Equivalent bending moment = $\frac{1}{2}$ M + $\frac{1}{2}\sqrt{M^2+T^2}$

M and T being in tons-inches.

Allowing an index figure for bending stress of not more than 13 tons per sq. inch gives the required minimum diameter of axle. The centre portion is made this diameter and the journals are increased by $\frac{1}{4}$" or $\frac{3}{8}$" in diameter to allow for wear. This method makes the assumption that all the torque from one cylinder is momentarily transmitted through the axle to the opposite side, which represents an artificial condition. Also, an allowance for the vertical bending stress due to the static load on the journals is included. The reactions are assumed to be concentrated at the centres of the journals.

Obviously, when the crank is vertical, the whole twisting moment cannot be transmitted to the other side, because at that moment the coupling rods on the other side are on the dead centre and cannot transmit any torque to the other axles. That leaves one driving wheel only to absorb the full torque, which is absurd. However, if the cranks are put at 45° to dead centre, the full torque of one cylinder *can* be transmitted to the other side, but its value is ·867 of the twisting moment taken. Then, if the vertical bending moment, due to static load, be combined with the horizontal bending moment, it will be found finally that stress arrived at is only about 4% less than by the other method.

The other coupled axles are not, of course, so highly stressed, but the bearings at least are generally the same size for all coupled axles. The centre portion may be reduced in diameter if desired, roughly on the basis of full piston load applied at the crank-pin.

The bearing pressure for coupled axles, taken on the projected area of the journal, is usually 180 to 200 lbs. per sq. inch, but for shunting tank locomotives this is sometimes as high as 240 lbs. per sq. inch.

Connecting Rods.

The usual bearing pressures allowed for connecting rods are:

Small end, 4000 to 4800 lbs. per sq. inch.
Big end, 1500 to 1900 ,, ,,

These figures are based on the projected area of the bearing in each case. The small end bearing should be of a length at least equal to its diameter. The body of the rod (if fluted) is somewhat less in width than the length of the small end bearing. If necessary, the length of bearing may be increased to give the most suitable section of rod.

The loads on a connecting rod are—

(1) *Full piston load*, to be taken in tension, and also in compression considered as a strut. Actually, the load is slightly more, due to angularity, but this is neglected.

(2) *Inertia stress*, from the vertical bending moment due to the weight of the rod itself when running at maximum speed, to which is added some direct stress due to the piston load corresponding to that speed.

For main line locomotives the rods are fluted to obtain rigidity with lightness, but for small shunting locomotives this is unnecessary. The body of the rod is tapered in depth by 1″ or more, with flanges of uniform thickness throughout when fluted.

The maximum tensile stress at the small end of the body of the rod is about 4 tons per sq. inch for Class C steel, which is the material usually employed. (B.S.S. 24/8C, 32-38 tons tensile).

For the section at the centre of the rod, consider first the loading as a strut, with full piston load at low speeds, when vertical loading due to inertia is negligible. In the vertical direction the rod forms a strut with hinged ends, while in the horizontal direction the ends may be considered as fixed, or they may be treated as hinged since the fixing is not rigid. In practice, both methods concerning this question of end conditions in the horizontal direction arrive at the same result for any normal rod, as the factors of safety allowed in each case are the outcome of experience over a long period of years.

Strut formulae in common use for rods are—

(1) *Euler's Formula* (which is strictly accurate only for very long columns),

$$\text{Buckling Load (lbs.)} = \frac{\pi^2 E I}{l^2}$$

where E = Modulus of elasticity $= 30 \times 10^6$ lbs. per sq. inch.

I = Moment of Inertia $\left\{ \begin{matrix} I_{xx} \text{ for vertical buckling.} \\ I_{yy} \text{ for horizontal buckling.} \end{matrix} \right\}$ (See Fig. 4).

l = Centres of rod in inches.

$$\text{The Factor of Safety} = \frac{\text{Buckling Load}}{\text{Working Load}}$$

It varies from 1·1 to 2·0 in the horizontal direction when considered as hinged ends. In the vertical direction the factor of safety is usually at least 6, considered as a strut only, depending somewhat on the speed.

(2) *Rankine-Gordon Formula*, in which constants are used for different materials, of which full particulars will be found in Low's "Applied Mechanics" and elsewhere.

$$\text{This gives the Buckling Load P (tons)} = \frac{A f}{1 + a\left(\frac{L}{R}\right)^2}$$

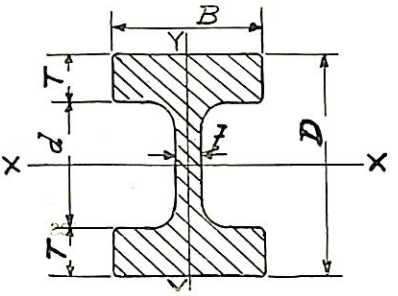

Fig. 4.

where f = ultimate crushing strength of material in tons per sq. inch.
A = area of cross-section in sq. inches.
L = centres of rod in inches.
R = radius of gyration of section in inches $= \sqrt{\frac{I}{A}}$
R_{xx} for vertical buckling.
R_{yy} for horizontal buckling.
a = a constant for end fixing and material.

For steel to B.S.S. 24/8, Class C, f = 32 tons per sq. inch and the constant a = about $\frac{1}{20{,}000}$ for fixed ends and $\frac{1}{5000}$ for hinged ends.

The factor of safety allowed with this formula and these constants may be 3·3 to 4·6 horizontally (fixed ends), and 4 to 5·5 vertically (hinged ends).

(3) *Merriman's Formula*, of American origin, gives

for hinged ends (vertical buckling) $f = \frac{10\ B\ E\ R^2}{10\ ER^2 - BL^2}$

and for flat ends (horizontal buckling) $f = \frac{10\ BER^2}{10\ ER^2 - \cdot 25\ BL^2}$

where f = stress in lbs. per sq. inch = 36,000 lbs. per sq. inch approx.
B = buckling load in lbs. per sq. inch of section.
L = centres of rod in inches.
E = modulus of elasticity = 30×10^6 lbs. per sq. inch.
R = radius of gyration, vertical or horizontal, as appropriate.
R_{xx} for vertical buckling.
R_{yy} for horizontal buckling.

Factor of Safety $= \frac{B}{\text{Working Load (lbs./sq. in.)}}$ = about 2 to 4

for flat ends (horizontal) and 2·5 to 4·5 for hinged ends (vertical), for steel to B.S.S. 24/8 Class C.

As an example, for a rod of 8′-0″ centres, whose moment of inertia $I_{YY} = 2{\cdot}13$ and $I_{XX} = 11{\cdot}2$, loaded with 20·5 tons, steel 32 tons breaking strength. Area of section at centre, 5·5 sq. ins. 16 tons yield point.

Factors of safety by each formula are—

Euler,	vertically, hinged ends,	7·83
	horizontally, taken as hinged ends,	1·49
	(5·96 if taken as fixed ends).	
Rankine-Gordon,	vertically, hinged ends,	4·52
	horizontally, fixed ends,	3·9
Merriman,	vertically, hinged ends,	2·8
	horizontally, flat ends,	2·52

Note that Merriman's factor of safety is based on the yield point of the steel, taken as 50% of breaking strength.

These formulae are rather confusing, and, generally speaking, the best thing to do is to use the one favoured in one's office, thereby maintaining a true basis of comparison with the firm's existing records, which is the most important thing after all. Euler's formula is the simplest to apply, but it should always be remembered that it is really applicable to a strut which is long in proportion to its cross-section, or, in other words, of which the ratio

$$\frac{\text{length}}{\text{radius of gyration}} = 80 \text{ or more,}$$

so that if used for very short rods a higher factor of safety should be allowed.

For the *Inertia Stress* in a connecting rod at high speed,

$$\text{Max. B.M. (in lbs.-inches)} = \frac{\cdot 768\, w\, v^2\, l^2}{g\, r}$$

where w = average weight of body of rod in lbs. per foot.
v = velocity of crankpin in feet per second.
l = centres of rod in feet.
r = crank radius in feet.

This maximum bending moment occurs at a distance of $\cdot 577\, l$ from the small end, but it is assumed to act at the centre of the length. The stress allowed for inertia alone is 3 to 4 tons per sq. inch for steel to B.S.S. 24/8 Class C, and may be increased in proportion when using special high-tensile steels. Note, however, that the use of such steels does *not* increase the safe load as a strut in proportion to the increase in ultimate strength of the material, and it is safer to treat it as though it were the usual B.S.S. 24/8 C. steel for the strut calculation, using the smallest factors of safety quoted.

Coupled with the vertical bending stress, due to inertia, there is also a direct stress in the rod due to the load being transmitted at

full speed, which may be taken as approximately 30% of full piston load.

To obtain the most suitable section for a fluted rod conformin with these various requirements, the following procedure is suggested :—

Fix a dimension for B (see Fig. 4), somewhat less than the length chosen provisionally for the small end bearing, and consider first the strength of the rod to resist buckling horizontally. From Euler's formula, using the factor of safety already stated, find the required Moment of Inertia = I_{YY}, assuming dimensions for d and t, which do not greatly affect this moment of inertia. Then

$$I_{YY} = \frac{2\,T \times B^3}{12} + \frac{d \times t^3}{12}$$

from which the thickness of flange T is obtained. Now, make the depth D such that the Moment of Inertia I_{XX} is about $5\frac{1}{2} \times I_{YY}$; work out the weight per foot for the section obtained, and test for vertical bending stress due to inertia. Adjust D to suit, and so work out the correct stresses and factors of safety, allowing for the fillets in the flutes. Also check the direct stress at small end of body of rod and thicken the web if necessary to suit this. If the section so found seems to have abnormally thick flanges compared with similar rods that is an indication that the dimension B should be increased and the length of the small end bearing altered to suit.

If Euler's formula be not favoured, the section may be tested according to the one in use. The procedure indicated obviates some blind guessing, and points the way to the most economical section.

The web thickness t is frequently as little as $\frac{3}{8}''$, especially when using special steels. Fillets of at least 1″ radius should be provided when using such a thin web.

The design of connecting rod ends varies greatly, depending on the type of bush, etc., which may be chosen. The maximum tensile stress at any point should not exceed that allowed for the body of the rod. It is generally somewhat less to allow for stress concentration round the bush, pressure applied in pressing in the bush, bending effect, etc. When solid ends are employed sufficient metal is required round the bush to ensure it remaining tight under load. Examination of current practice shows that for the section through the end of the rod 4 to 5 tons of piston load per square inch of metal is suitable. Changes of section should not be abrupt.

Where strap ends are used, or solid rectangular eyes for adjustable bushes, ample material should be allowed in the end to avoid deflection and consequent distortion of the eye.

Coupling Rods.

These are treated on similar lines to connecting rods, with the following exceptions.

The maximum bending moment on a coupling rod due to inertia, since its motion is entirely rotary, is

$$\text{B.M.} = \frac{1{\cdot}5\ w\ v^2\ l^2}{g\ r} \text{ (lbs. inches).}$$

the symbols being the same as given for connecting rods.

The load to be allowed for when considered as a strut is not so clearly defined as in the case of the connecting rod. Certain wheels may slip momentarily and throw an extra load on some sections of the coupling rod. Also, it is difficult to estimate the effect of injudicious sanding of rails with a locomotive slipping at full throttle. Misalignment, due to excessive side-play of wheels with badly-worn axle-boxes, might well be a cause of failure. Experience shows that for a four-coupled or six-coupled locomotive it is desirable to assume that the whole piston load might be transmitted by any one section of the coupling rod, and a factor of safety similar to that for the connecting rod is allowed. For the outer section of the coupling rod of an eight-coupled locomotive, full piston load is asking too much of the adhesive grip of the wheel on the rail, so a compromise is made by allowing for about three-quarters of the piston load. As already mentioned, offsets should be avoided in coupling rods.

Joint pins between sections may be plain pins, or pins with a taper head in one jaw and a split taper bush in the other, similar to those sometimes used in crossheads. The bearing pressure allowed on these pins is about 2500 lbs. per sq. inch taken on normal loading as for crankpin bearing pressures.

Coupling rods are commonly fluted, though for short rods it is the direct stress which counts most and consequently a plain rectangular section is quite suitable, besides being a lighter forging and costing less to machine. For express locomotives with long coupling rods of 10 ft. centres or more (as on some four-coupled locomotives), great care in designing is required, as it is possible, when trying to increase the strength of the rod vertically, to add material in such a way that the weight per foot is increased in greater proportion than the section modulus, with the result that although the rod is stronger than it was before, the stress due to inertia is actually increased. Trial with various sections is essential in such a case in order to find the most suitable section. Such long coupling rods are usually deeper at the centre than at the ends, with the fluting parallel throughout, thus giving thicker flanges in the centre where the maximum stress occurs.

The leading end of a coupling rod frequently passes behind the crosshead in its path, and a clearance of at least $\frac{3}{4}$" should be allowed. To obtain this clearance without increasing the width between cylinders, the leading crankpin is frequently fitted with a sunk cap instead of the nut arrangement usually employed.

Floating bushes on connecting and coupling rods are sometimes

fitted. Floating bushes are usually of cast iron, pierced with holes, and grease lubrication is employed. They have a longer life than ordinary bushes, but the size and weight of the rod end is greater.

Roller bearings are also being experimented with, and there again an increase in weight is unavoidable.

Crossheads.

Crossheads are of many types, and their detail construction varies enormously. It is proposed to discuss here only the main points in their design.

Gudgeon Pin.—The diameter and length of bearing for the connecting rod small end are fixed by reference to the bearing pressure and the width of the connecting rod body. It will be found that shear and bending stresses in the pin are small. The pin is tapered to engage the crosshead and is accurately fitted. Fig. 5 shows a solid pin and also a pin with a split tapered bush at one end. A taper of 1 in 6 is common, but this varies greatly. For the bearing pressure on the slipper, the maximum load occurs when the crank is vertical, but it is taken for convenience as

$$\text{Piston Load} \times \frac{r}{l} \quad \text{where } r = \text{crank radius}$$

and l = centres of connecting rod.

The error in this is negligible.

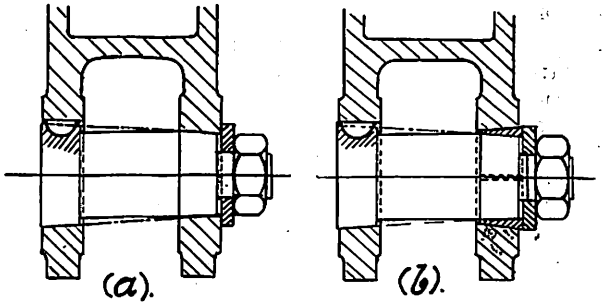

Fig. 5.

Sometimes the centre line of the cylinder is horizontal but raised a distance $h = 1''$ to $2''$ above the centre line of the axles, in which case the maximum bearing load on the crosshead slipper occurs with the crank down and is equal to Piston Load $\times \frac{(r+h)}{l}$.

The bearing pressure allowed is about 60 to 80 lbs. per sq. inch for main line locomotives, while for shunting locomotives on intermittent work it is sometimes as high as 90 lbs. per sq. inch. The

length of slipper is generally 4 to 5 times the width, depending somewhat on coupling rod clearances, etc. Crossheads are most conveniently made of cast steel, with renewable slippers of gunmetal, or of cast iron with white metal inserts.

For the cotter joint to the piston rod the following conditions must be satisfied simultaneously :—

(1) Tensile stress in the piston rod across the cotter hole to be $4\frac{1}{2}$ to $5\frac{1}{2}$ tons per sq. inch.
(2) Shear stress in the cotter (double shear) to be also about $4\frac{1}{2}$ to $5\frac{1}{2}$ tons per sq. inch.
(3) Bearing pressure on the cotter to be about 8 to 9 tons per sq. inch.

The following procedure is suggested—

If P = piston load in tons.
f_t = tensile stress across cotter-hole in tons per sq. inch.
f_s = shear stress in cotter in tons per sq. inch.
f_b = bearing pressure on cotter in tons per sq. inch.
d = diameter of piston rod at cotter hole.

It can be shown that $d = \sqrt{\dfrac{4\ P\ (f_b + f_t)}{\pi \times f_b \times f_t}}$

If f_t and f_s are each made $= 5\frac{1}{2}$ tons per sq. inch and $f_b = 9$ tons per sq. inch,

then $d = \sqrt{\cdot 373\ P}$

Also, with the same allowable stresses,

Thickness of cotter $t = \dfrac{\pi \times d \times f_t}{4\ (f_b + f_t)}$

which reduces to $t = \cdot 298\ d$.

Taking the nearest suitable dimensions for d and t, the breadth of the cotter and the actual stresses are easily arrived at. The end of the piston rod in the crosshead is usually tapered about 1 in 12 to 1 in 20.

The section of material between the cotter hole and the end of the piston rod is subject to a crushing load. For a round-edged cotter, this is assumed to be concentrated on the centre line, and a stress of 4 to 5 tons per sq. inch is allowed. A similar stress is allowed through the crosshead neck in front of the cotter, the diameter of the neck being increased at the end to give the required bearing surface for the cotter.

The vertical height from the centre line of the cylinder to the slide-bar is made just sufficient to accommodate the small end of the connecting rod, so that the body of the connecting rod is just clear of the slide bar at all points of the stroke under the worst conditions, *i.e.*, with the axlebox up to its stop and the axlebox brasses worn.

It may be mentioned here that sufficient clearance must be allowed between the crosshead neck and the piston rod swab box, and between the crosshead slipper and the slide-bar bracket at the other end, to allow the piston to be bumped up against the cylinder covers when erecting, so that the actual clearances of the piston in its working stroke may be measured. In addition, some railways require the clearance at the swab box end to be such that the piston can come out of the cylinder at the front end far enough to be able to renew the piston rings without having to uncouple the crosshead from the piston rod. This is a good feature from the maintenance point of view, but it reduces the possible length of the connecting rod, which is undesirable if the rod happens to be very short already.

It is important to see that ample material is provided in the neck and jaws of all types of crossheads to transmit the full piston load, and that the whole crosshead is of a rigid shape. Also, it is obvious that it should be as light as possible, as it forms a large part of the reciprocating weight.

Slide-Bars.

The load on a slide-bar is a moving load of varying intensity, distributed over the length of the crosshead slipper. In practice, however, it is convenient to consider the load as concentrated, and on that assumption to allow a bending stress of 4·5 to 5 tons per sq. inch, with a corresponding deflection of about ·015″ per foot of of length between supports. These figures represent a convenient basis of comparison with other slide-bars rather than accurate statements of fact. The maximum load occurs when the crank is vertical (see under crossheads). If this load occurs when the crosshead is at or near the mid-distance between the slide-bar supports the maximum bending moment is at this point, and the calculation is merely that for a simply supported beam with a single load. It frequently happens with double slide-bars that the bars project some distance beyond the bracket, in which case the load at various points of the stroke must be worked out, and trials made to find which gives the maximum bending moment.

The general expression for the vertical load on a slide-bar (see Fig. 6) is

$$\text{Load on slide-bar, } W = P \tan \theta.$$

Sin $\theta = \frac{x}{l}$, and since θ is a small angle tan θ = sin θ approximately, hence for our comparative basis of calculation it is sufficiently accurate to take load on slide-bar, $W = P \times \frac{x}{l}$, which is zero at dead centre and a maximum when x = crank radius.

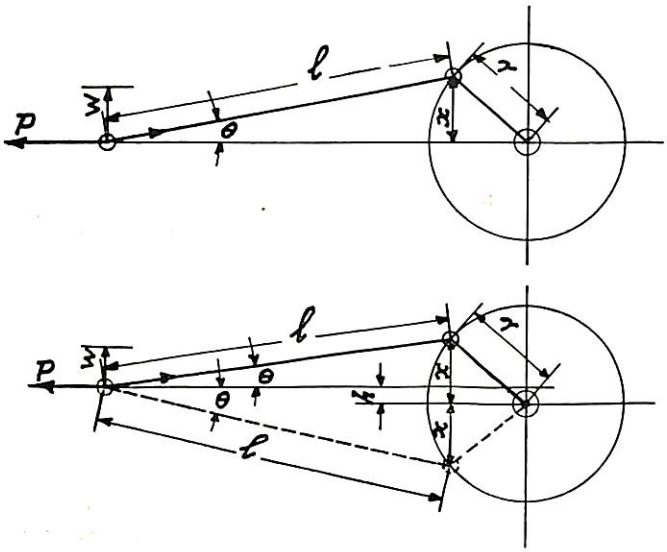

Fig. 6.

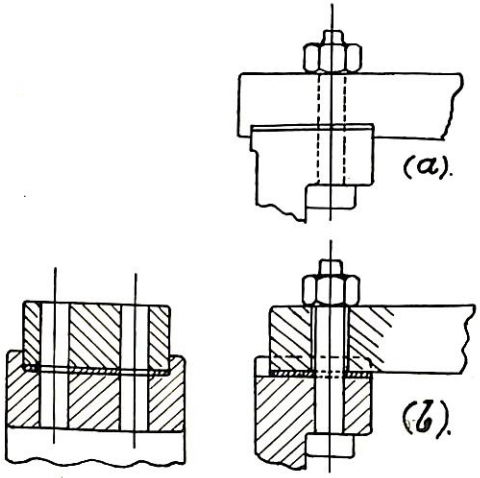

Fig. 7.

When the cylinder centre line is a distance h above the centre line of the axles,

$$W = P \times \frac{(x + h)}{l} \text{ for one stroke,}$$

$$\text{and} \quad P \times \frac{(x - h)}{l} \text{ for the other stroke.}$$

For forward running, W acts upwards always, and vice versa.

The fixing of the slide-bar to the cylinder cover should be with bolts if possible, which are stressed low, about 1·5 tons per sq. inch at root of thread. Many railways have the slide-bar lipped over the cover facing, as shown at (a) (Fig. 7) and located transversely by fitted bolts. On others the bolt holes in the slide-bar are elongated to allow for expansion (b) (Fig. 7), and a machined lug is provided on the cover, on each side of the slide-bar, to maintain true alignment. At the slide-bar bracket fitted bolts are used, with or without locating lugs. Where studs are fitted at the cylinder cover end a difficulty in assembly arises and taper holes may be necessary. Liners are fitted at each end so that wear in the crosshead slipper can be taken up. The crosshead slipper should overrun a rebate in the slide-bar at each end to avoid forming a ridge.

For single-bar crossheads the slide-bar should be deep to give stability to the crosshead. It may be fluted on the sides to reduce weight or alternatively it may be made generous in width if clearances permit.

Fig. 8 shows four types of crossheads and slide-bars in common use :—

(*a*) Ordinary double-bar type.
(*b*) Single-bar type with box crosshead.
(*c*) Twin single-bar type, presumably designed originally for a tender engine running mostly forward, as the top bar presents a greater surface than the bottom.
(*d*) Four-bar type with slide-blocks, formerly the standard type for inside cylinders, but now frequently replaced by a single-bar type.

Piston Rods.

The required diameter of a piston rod across the cotter hole has already been discussed (see Crossheads). The body diameter of the rod should be about $\frac{1}{8}''$ greater than the top of the taper in the crosshead, thus allowing for re-grinding. Sometimes the body is smaller than the tapered end, which saves weight but necessitates a split metallic packing. Where the body has been so reduced, or where the piston rod is abnormally long for some reason, it should be checked as a strut, assuming hinged ends, and should have a factor of safety of 8 at least.

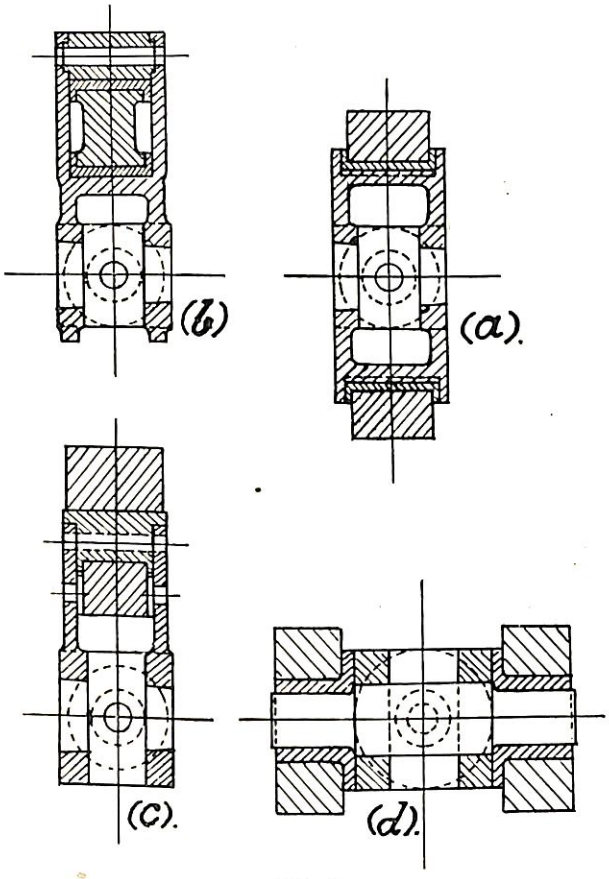

Fig. 8.

The end of the rod in the piston is usually tapered about 1 in 6, with or without a shoulder on the rod, and is secured by a gunmetal nut and a cotter. The screw is usually of such a diameter that the area at bottom of thread is about equal to that across the cotter hole in the crosshead, the tapered part in the piston being enlarged to suit. Another system, now used on Indian Standard Locomotives, has the piston rod screwed into the piston with a slow taper thread, and is locked with a grub screw, thus giving a flush finish. (The piston in this case has a cast steel centre).

It should be noted that the body of a piston rod is subject to alternating stresses, whereas the screwed end and the cottered end are subject to tension only. It is recommended that for ordinary carbon steels the stress in the body should be based on the elastic limit of the material and not on the ultimate strength. Thus, for

a piston rod of 40-45 ton steel with an elastic limit of 20 tons per sq. inch, a stress of $5\frac{1}{2}$ tons per sq. inch across the cotter hole and at the root of the thread respectively would give a minimum factor of safety on the *ultimate strength* of 7·3 approximately, while a stress of $2\frac{3}{4}$ tons per sq. inch in the body would give a factor of safety of 7·3 on the *elastic limit.* These figures refer to direct stress only, as distinct from stresses obtained from certain strut formulae.

Tail rods on pistons are not much used for cylinders of less than 20″ diameter, and even above that size they are frequently dispensed with. The tail end may be enclosed in a steam-tight casing, or it may pass into the open through a standard type of metallic packing, the latter being the usual practice on the Continent. In either case —since a metallic packing is floating and supports none of the weight of the piston and rod—a gunmetal guide bush of ample length should be provided, and made reversible to take up wear, as the weight is always on the bottom. Alternatively, a small crosshead and slide-bar may be fitted in front of the cylinder, as on some recent Argentine locomotives.

The diameter of the tail rod must be such that the deflection due to the weight of the piston and of the rod itself is small enough to maintain a working clearance between the body of the piston and the cylinder barrel. Usually, the diameter is made about ·6 to ·75 of the body of the piston rod. The higher value applies particularly where the small slide-bar type of guide is used.

Since the function of a tail rod is to reduce the wear on the bottom of the cylinder barrel, much depends on periodical and careful adjustment at slide-bar and tail rod to preserve true alignment.

For the bearing surface of the guide an area of 25 sq. inches has been suggested as desirable. With a $2\frac{1}{2}$″ tail rod this would mean a bush 10″ long, which is much more than is usually practicable.

Walschaert's Valve Gear.

Fig. 13 shows diagrammatic arrangements of Inside Admission and Outside Admission valve gears. It also gives the nomenclature of component parts for reference.

The form and dimensions of valve gear are influenced by the general design of the frame, etc., and hence the relative lengths of various component parts vary considerably on different jobs. In text-books on valve gears one sometimes encounters formulae for the throw of the eccentric crank, etc., without indicating how they are arrived at. The writer considers it best to work the whole thing out from basic principles.

Taking some actual figures to illustrate the procedure, suppose it is required to provide a valve travel in full gear of $5\frac{1}{4}$″, with an *inside-admission* piston valve having a steam lap of $1\frac{1}{4}$″, port opening $1\frac{3}{8}$″, and lead $\frac{1}{8}$″, these dimensions giving the desired cut-off of 75% (see Fig. 3). The piston stroke is 20″

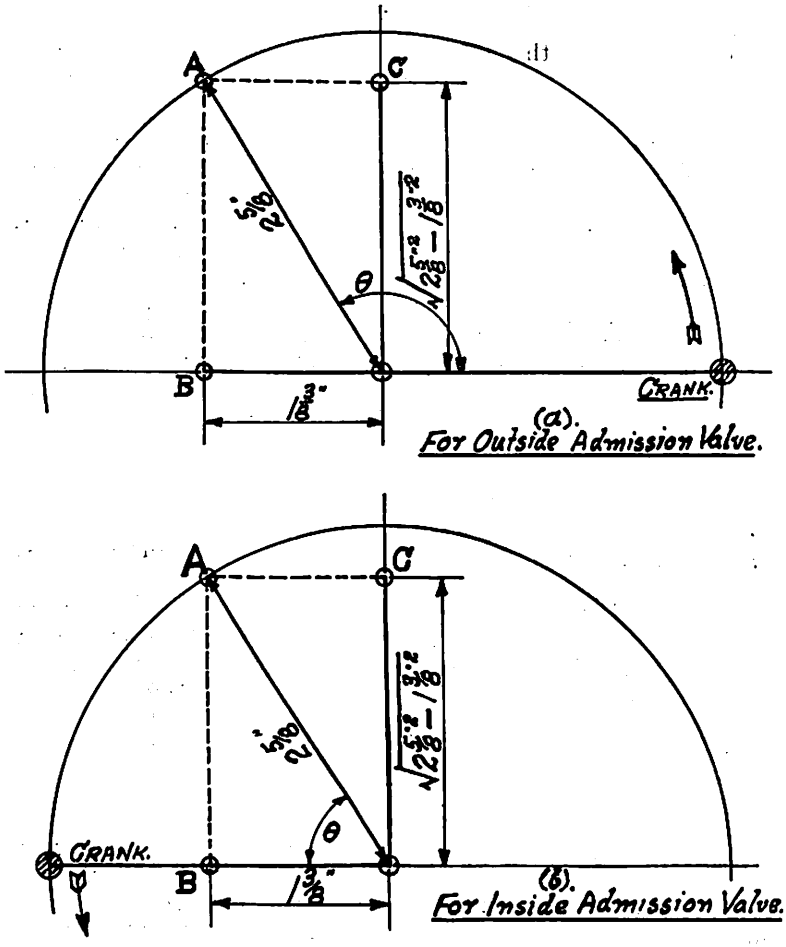

Fig. 9.—Equivalent Eccentric.

These conditions would be fulfilled by an eccentric (see Fig. 9) having a total throw of $5\frac{1}{4}''$ and set at an angle to the crank such that when the crank is on the dead centre the eccentric has moved a distance of (lap + lead) from its central position—$1\frac{3}{8}''$ in this case. This is called the "equivalent eccentric," and is represented by OA in Fig. 9.

This eccentric is split into two components, one OB in phase with the crank and one OC at right angles to it. OB is derived from the crosshead, through the combination lever, and OC from the eccentric crank through the reversing link, radius rod, and combination lever.

OB = lap + lead = $1\frac{3}{8}''$ half travel or $2\frac{3}{4}''$ full travel.

OC = $\sqrt{(OA)^2-(OB)^2} = \sqrt{2\frac{5}{8}^2-1\frac{3}{8}^2}$ = 2·236″ half travel or 4·472″ full travel.

Each of these represents travel at the valve spindle, considered independently of each other, and their combined effect when acting together is to produce a nett valve travel of $5\frac{1}{4}''$.

The combination lever normally has its bottom pin some distance below the crosshead centre, though in special cases it may be higher. The ratio of length (see Fig. 10) is such that

$$\text{Travel at valve spindle T} = 2\,(\text{lap} + \text{lead}) = \text{piston stroke} \times \frac{x}{l}$$

$$\therefore \quad l = \frac{\text{Piston stroke} \times x}{2\,(\text{lap} + \text{lead})}$$

The length x depends on the size of motion pins and the type of radius rod end. Where possible, the design shown by (a), Fig. 11, should be used, with a jaw on the combination lever only, this being easiest for manufacture. If this results in l being excessively long, the design may be as shown by (b) (Fig. 11), with a jaw on the radius rod also. In our example, suppose the motion pins are $1\frac{3}{4}''$ diam., then to contain the two rod ends as stated, x must be about 4″.

$$\text{Hence } l = \frac{20'' \times 4''}{2\,(1\frac{1}{4}'' + \frac{1}{8}'')} = 29{\cdot}091''$$

Note that the combination lever should not swing more than 60° in its full travel, *i.e.* l must be at least equal to the piston stroke.

The reversing link should be located about midway between the valve spindle end and the driving axle, if it is possible to arrange suitable brackets to carry it there. This gives the best working gear, but frequently it is impracticable, as inspection of existing locomotives will show. The point is not a vital one, so long as the reversing link is not crushed up unduly towards one end or the other.

Having fixed a position for the reversing link and thereby determined its radius (which is struck from the mid-position of the

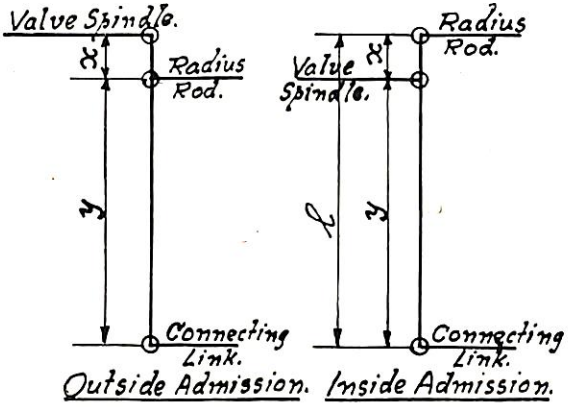
Valve Spindle.
x
Radius Rod.
y
Connecting Link.
Outside Admission.
Radius Rod.
Valve Spindle.
x
l
y
Connecting Link.
Inside Admission.

Fig. 10.

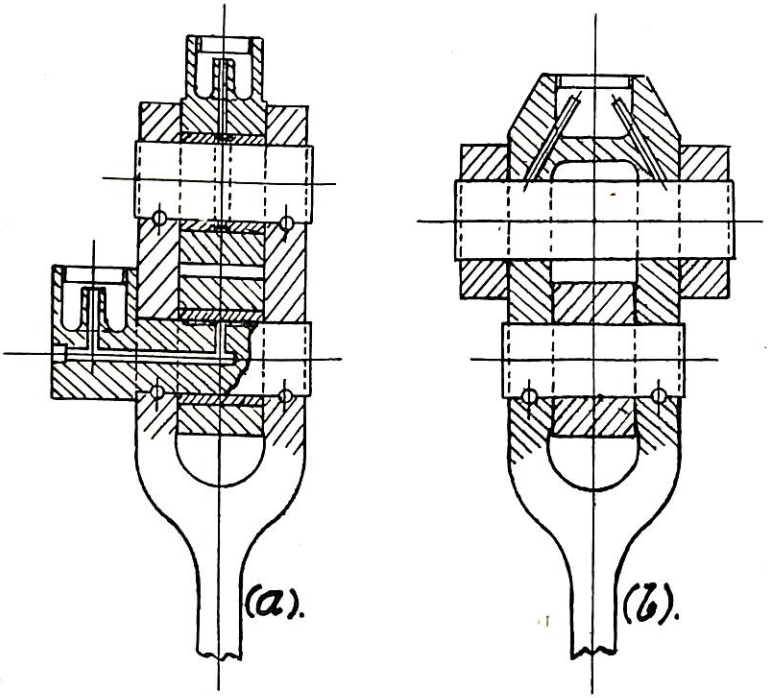
(a).
(b).

Fig. 11.

combination lever), the travel of the die-block in the reversing link on each side of its mid-position should be made approximately equal to the valve travel, or slightly greater if found necessary. It has been stated that for short radius rods this depth in gear should not exceed one-eighth of the length of the radius rod. In our example, suppose depth in gear $= 1\frac{1}{4} \times$ valve travel $= 6\frac{1}{2}''$.

The tail of the reversing link should extend well down towards the centre line of the axles. The correct theoretical position for this point is on the centre line of the axles, but in most cases the tail cannot be carried down as far as this without making the depth in gear and the throw of the return crank abnormally large.

In our example, suppose this length $= 14''$. To find the required throw of the eccentric crank, work back through the leverage system from the valve spindle, assuming the combination lever to be pivoted on its bottom pin, to find a throw of eccentric crank which would give the valve spindle a travel $= 2 \times$ OC (see Fig. 9). In our example $2 \times \text{OC} = 2 \times 2{\cdot}236 = 4{\cdot}472''$.

Thus (see Fig. 13),

$$4{\cdot}472'' = \text{throw of eccentric crank} \times \frac{6\frac{1}{2}''}{14''} \times \frac{25{\cdot}091''}{29{\cdot}091''}$$

hence throw of eccentric crank

$$= \frac{4{\cdot}472 \times 14 \times 29{\cdot}091}{6{\cdot}5 \times 25{\cdot}091} = 10{\cdot}98'', \text{ say } 11''$$

The valve gear can now be laid down to these dimensions, and it will be found finally that only a slight adjustment of the depth in gear will be required to obtain the correct valve events.

The height of the reversing link trunnion centre is frequently the same as the centre line of the piston valve, and should not normally be lower than this. The radius for the mid-position of the reversing link is struck from the pin connecting the radius rod to the combination lever, with the crank on dead centre and the valve moved a distance = (lap + lead) from its mid position. It follows from this that with the crank on either dead centre the die-block can be moved through its full range in the reversing link without moving the valve, *i.e.*, the lead is constant under all circumstances. This test must be applied when setting up the gear on a working model or on the locomotive itself.

The tail of the reversing link lies on a radius of 14" (in our example) from the trunnion centre, and its correct position must be found by trial. Assume some point on the arc of its travel, and draw a line from it passing through the centre of the driving axle, and another line at right-angles to that intersecting the eccentric crank circle (see Fig. 12). Join AB, and with AB as radius scribe from C and D to find the limits of travel of the point A. If this travel is found to be unequal about the mid-position A, repeat from some other point until equal travel is obtained.

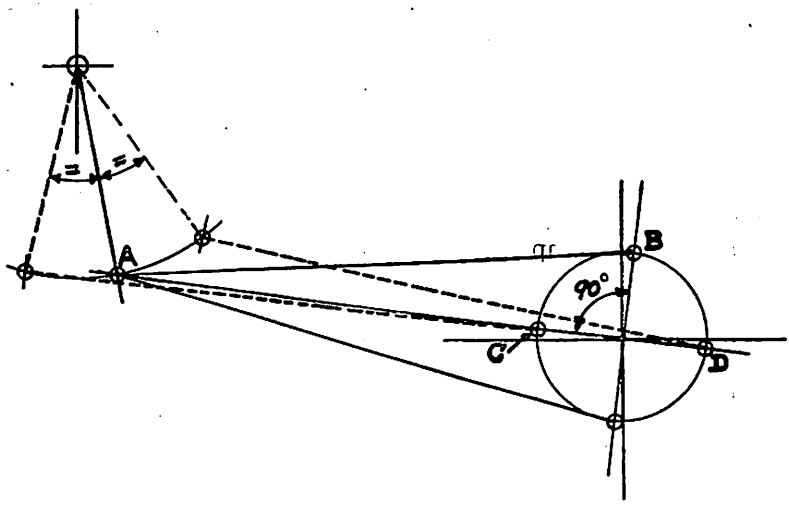

Fig. 12

The reversing shaft remains to be fixed. This may be placed before or behind the reversing link, and may carry the radius rod on a lifting link, or on a die-block sliding in a slot in the radius rod. Only when carried on a lifting link is it possible to influence the valve events by varying the position of the reversing shaft. Judging from the fact that the latest standard valve gears on the L.M.S. railway and the L.N.E. railway are of the sliding block type, this point cannot be regarded as of great importance.

With the suspended link arrangement, the procedure for locating the reversing shaft is as follows.

Choose a suitable point on the radius rod for attachment of the lifting link, as near the die-block as will ensure a working clearance between the lifting link and the top of the reversing link in all positions. Also assume a length for the lifting link, making it as long as conditions will allow. Taking full forward gear, set down (centre lines only) the crosshead centre at the desired cut-off percentage of stroke, the connecting rod, giving the corresponding position of the crank; from that the corresponding eccentric crank position; then the eccentric rod, giving the centre line of the reversing link at the point of cut-off. Set the valve at cut-off point, and from the crosshead and connecting link draw the corresponding position of the combination lever. Then an arc struck from the combination lever with radius = length of radius rod, intersecting the centre line of the reversing link already drawn gives the corresponding position of the radius rod. On this line the point chosen for attachment of the lifting link is marked. Repeat the procedure for the same cut-off percentage of the return stroke, giving a second point for the bottom of the lifting link. Arcs struck from these

two points with radius = length of lifting link give a position for the top of the lifting link corresponding to equal cut-off on both strokes.

Repeat the whole procedure for full backward gear and for at least one intermediate cut-off in forward and backward, giving a locus of points for the path of the reversing shaft arm. It remains then to choose a position for the centre of the shaft and a radius of arm which will conform as nearly as possible to these points. This procedure is too complicated for a small illustration. In practice it is quite simple if followed through systematically as described.

The arrangement drawing of the motion may now be completed, and particular attention must be paid to the actual clearances of all moving parts.

Finally, the actual performance of the gear is required. This may be deduced on the drawing board, given time and patience, by taking ten or twelve divisions on the stroke, finding corresponding positions for the crank and the reversing link, and then using a template for the combination lever and the radius rod to find the position of the valve corresponding to each division of the stroke. The valve position is then plotted on a diagram against the stroke, giving the "valve ellipse," and a line drawn on this for the lap of the valve gives the points of cut-off, release, etc., and the port opening. This whole procedure is necessary for each notch in the gear if full data be required. It is a tedious business, while the results obtained are none too accurate, as it is seldom possible to work to a scale larger than quarter full size. For this reason some firms have a full-size skeleton model on which any gear may be accurately assembled and the full performance read off with ease. Such a model can draw the valve ellipse automatically, if desired. Its use makes it easy to study the effect of varying the position of the reversing shaft, and also to measure the slip of the die-block in the reversing link. The amount of this slip is important, and should not exceed 1" if possible, at any time, as slip means friction and wear of die-block.

On a locomotive with a short connecting rod the crank positions corresponding to 75% of the outward and inward strokes are not nearly at 180° to each other, and equal cut-off in both strokes can only be attained at the expense of increased slip of the die-block. A reasonable compromise must therefore be made. Also, if with equal cut-off the port openings should prove to be very unequal, a compromise is again indicated.

The other type of reversing shaft, where the arm carries a die-block which slides in a slot in the end of the radius rod, is much favoured on many railways in spite of the fact that it requires an expensive "box" type of reversing link. A strong point in its favour is the greater leverage it gives on the radius rod, making the locomotive easier to reverse. Also, less "kick" is transmitted back

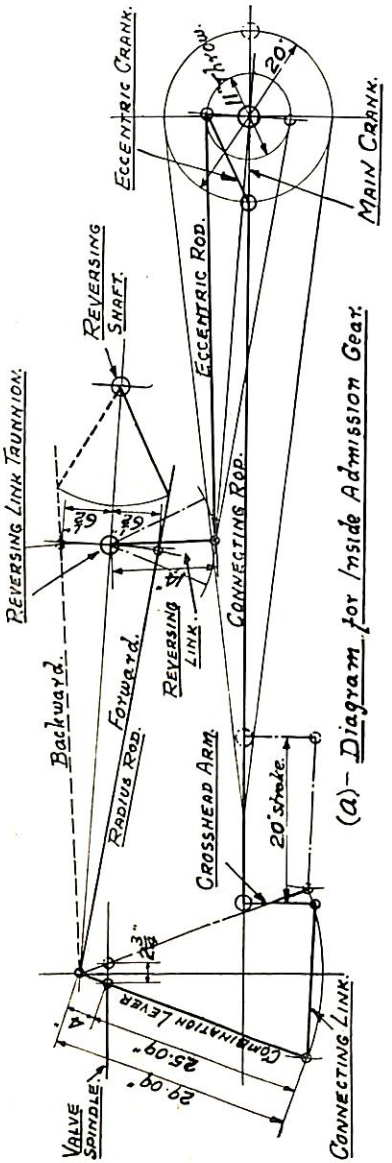
Reversing Link Trunnion.
Backward.
Forward.
Radius Rod.
Reversing Shaft.
Reversing Link.
Crosshead Arm.
Eccentric Crank.
Eccentric Rod.
Connecting Rod.
Main Crank.
20° stroke.
Valve Spindle.
Combination Lever
Connecting Link.
(a) - Diagram for Inside Admission Gear.
Backward.
Forward.
(b) - Diagram for Outside Admission Gear.

Fig. 13.

through the reversing gear. (This "kick" is the vertical component of the load being transmitted to the radius rod from the reversing link during its swing, being a maximum at each end of the swing and zero at the mid-point). Also, the slip of the die-block when working in the top of the link is less than with the suspended link arrangement. On the other hand, it is not possible to make corrections in the valve performance by varying the position of the reversing shaft, and for that reason this type of reversing should not be applied to engines having very short connecting rods, if at the same time an equal performance for outward and inward strokes is being aimed at.

In the diagrams shown for inside admission valve gear, forward gear is shown with the die-block in the bottom of the link, and for that arrangement the eccentric crank follows the main crank when running forward. If the forward gear is desired to be with the die-block in the top of the link, the eccentric crank is moved through 180° and leads the main crank when running forward.

All the foregoing notes refer to valve gear for *Inside Admission* piston valves. When the valve gear is required to operate *Outside Admission* piston valves, the treatment is on exactly the same lines, except for the following points of difference which emerge (see Fig. 13 b).

The radius rod is connected to a point on the combination lever *below* the valve spindle.

The ratio of length of the combination lever (see Fig. 10) is such that

$$\text{Piston stroke} \times \frac{x}{y} = 2\ (\text{lap} + \text{lead}).$$

The throw of the eccentric crank (see Fig. 13),

$$T = \frac{2 \times OC \times y \times c}{l \times d}$$

in which OC is as shown on Fig. 9, being the same as for Inside Admission.

The eccentric crank leads the main crank when running forward, with the die-block in the bottom of the link in forward gear. It follows the main crank if forward gear is at the top of the link.

Note that the eccentric crank position in relation to the main crank is the converse of that pertaining to an Inside Admission valve.

Mention has already been made of locomotives designed with the centre line of the cylinders horizontal, but raised some distance above the centre of the driving axle. In designing the valve gear for such a locomotive a slight difficulty arises from the fact that there is no true dead centre of the crank. It is usual to draw a line joining the mid-point of the crosshead stroke and the centre of the driving axle, and to take the points where this intersects the crank circle to be the dead centres. (See Fig. 14).

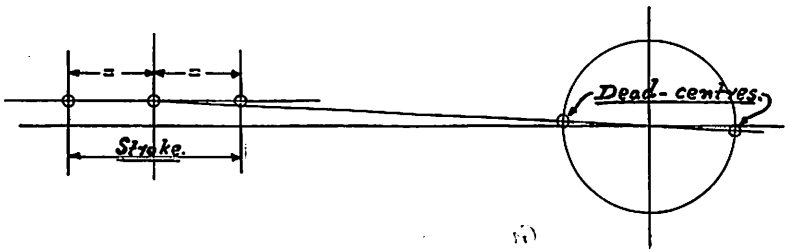

Fig. 14.

Valve Gear Details.

Valve gear details must be of ample section and rigid design. They should be as light as these considerations will permit, since their weight cannot be properly balanced. Unfortunately, the force required to overcome the friction of a piston valve cannot be ascertained with any certainty, and in addition the force required to accelerate a piston valve is difficult to calculate. While it is possible to estimate these forces by making various assumptions, it is much better (and easier) to base consideration on some similar existing gear which is known to be of good proportions, and to make allowances for difference in lengths of rods, etc. Broadly speaking, the required strength of valve gear is governed by the size and weight of the piston valve, the maximum speed, and to some extent by the steam pressure (except where the valve has special floating rings).

It must be remembered, also, that quite a number of existing valve gears were originally designed for working slide valves, and are heavier than is really necessary for piston valves.

The diameter of the motion pins bears a relation to the piston valve diameter somewhat as follows :—

Piston Valve Diameter,	8″	9″	10″	12″
Motion Pin Diameter,	1½″	1⅝″	1¾″	2″

The bearing length of the pin should be at least equal to its diameter.

The pins and wearing surfaces are case-hardened. Steel or bronze bushes ¼″ thick may be fitted in the end which moves relatively to the pin. If not bushed, allowance must be made for re-boring the holes to a larger diameter when wear has taken place. Recently, special needle roller bearings have been applied successfully to motion pins.

The eccentric crank bearing, according to existing practice, is made about 2½″ diameter for 8″ valves, 2¾″ for 10″ valves, and about 3″ diameter for 12″ valves, with length not less than diameter. A gunmetal bush is pressed in the rod and fixed by a key or set screw. A renewable steel bush is sometimes fixed on the eccentric crank to form the bearing.

Alternatively, a self-aligning ball bearing may be fitted.

Eccentric rods and radius rods, if long, may be fluted to obtain rigidity with minimum weight. Reversing links should be carried on two trunnion bearings, with renewable gunmetal bushes. Alternatively, needle roller bearings may be fitted.

Overhung pins in valve gear should be avoided, but if used they should be fitted tightly with a long seating and riveted over or secured by a nut.

Valve Spindle Guides are of two main types—a plain circular bearing of about $3\frac{1}{2}''$ diameter or more, fitted with a gunmetal bush, or an arrangement comprising two slide-blocks on the ends of the valve spindle pin working in guides. The latter type is more expensive, but is superior as regards lateral stability. Usually a set screw or some such provision is made whereby the valve may be locked in its mid-position with the ports covered in the event of a rod breaking, thus shutting off the steam from one cylinder.

Lubrication.

When oil lubrication is used, syphon oil reservoirs are formed from the solid on the rods, etc., with the exception of the combination lever, where it is sometimes convenient to form the reservoir on the ends of the pins. (See Fig. 11). Some railways use grease for lubrication, forcing it in through nipples on the ends of the pins. When the rods are bushed and the oil hole passes through the bush, a groove should be turned on the outside of the bush to ensure an oil supply even if the bush should turn slightly in service. (See Fig. 11).

Three-Cylinder Locomotives.

This paper has, so far, been concerned exclusively with the design of two-cylinder simple locomotives with outside cylinders.

It is now proposed to deal briefly with some important aspects of a three-cylinder design.

Suppose we have a three-cylinder simple locomotive with cranks at 120° and all driving the same axle. The piston load on each cylinder = 20 tons and crank radius = 10″ = R. Three coupled axles. (See Fig. 15a).

The maximum torque on the driving axle may be obtained when the L.H. crank is on dead centre and the R.H. and centre cranks are both taking full piston load,

$$\text{Max. torque} = 20 \text{ tons} \times x \times 2.$$
$$x = R \sin 60° = \cdot 867\ R$$
$$\therefore \text{ Max. torque} = 2 \text{ (piston load on one cylinder} \times \cdot 867\ R)$$
$$= 2 \times 20 \text{ tons} \times \cdot 867 \times 10 = 346 \cdot 8 \text{ tons inches.}$$

This torque is absorbed equally by each axle,

$$\therefore \text{ Torque transmitted by coupling rods} = 346 \cdot 8 \times \tfrac{2}{3} = 231 \cdot 2 \text{ tons inches.}$$

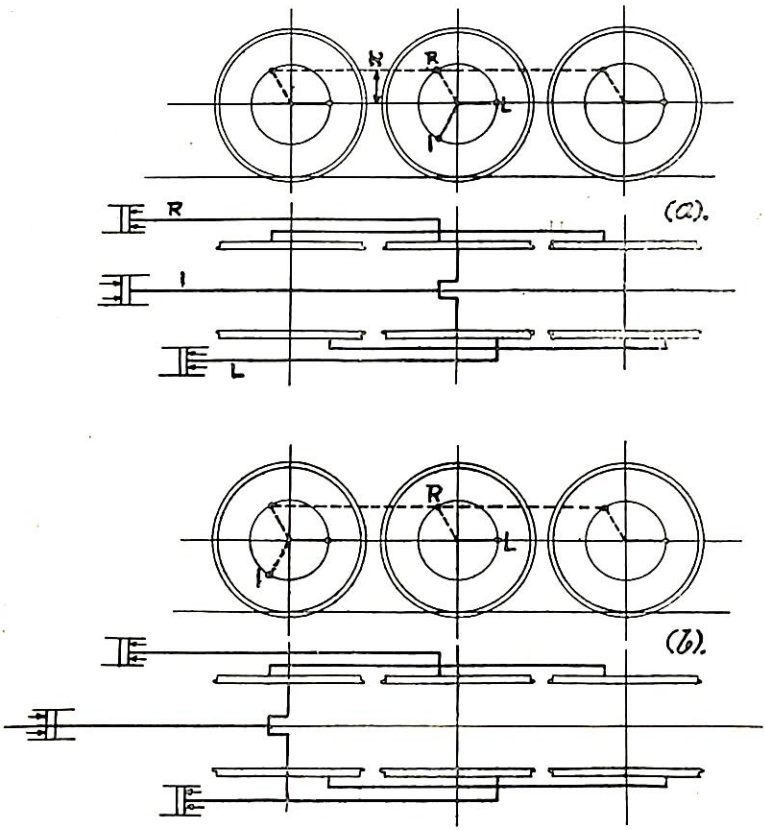

Fig. 15.

Since the L.H. crank is on the dead centre, all this is taken by the R.H. coupling rods. For the arrangement shown, each half of the R.H. transmits half of this torque, and the load on each half of the coupling rod is given by

$$\frac{231{\cdot}2}{2 \times {\cdot}867 \times 10} = 13{\cdot}33 \text{ tons.}$$

This is the bearing load on the leading and trailing crank-pins, on which the bearing pressures should be based. The coupling rod bearing load on the driving crank-pin is, of course, the sum of that on the leading and trailing.

For the strength of the coupling rods, it might be assumed as before that the trailing wheels slip, and the leading half of the coupling rod takes double the load. It would be more correct to say that trailing wheels slip and *half the total torque* is transmitted

by the leading coupling rod. An examination of coupling rod sections on some well-known three-cylinder engines on British railways shows that the latter basis conforms best to what has been found suitable in practice.

If all three cylinders drive on the leading axle, the coupling rod bearing loads are unaltered, except that the leading crank-pin now takes 26·66 tons and the other two 13·33 tons each. For the strength of the coupling rods in this case, the leading half must be capable of transmitting the full torque if the leading wheels slip, giving a load of 40 tons in the example considered, while the trailing half will take at least half of this.

These loads represent a simple basis of design and comparison, and are no doubt open to criticism. It may be argued that one should derive the maximum loads on coupling rods by assuming a coefficient of friction at which the wheels will slip, but this coefficient is variable and not known with any exactitude. The question of variation in rail loads should also be considered with it, so it does not seem as though this basis of design can be any more reliable than the one given here.

The drive may be split, with the centre cylinder driving the leading axle and the two outside cylinders driving the second axle. (See Fig. 15b).

With this arrangement the maximum stresses in the coupling rods are attained when the L.H. crank is on the dead centre as shown, but the bearing loads should be taken with the R.H. crank vertical, and the centre crank assumed to be unloaded, since its position corresponds to about 92% of stroke, which is at, or beyond, the exhaust point.

Thus for *Bearing Loads.* With the R.H. crank vertical,

Total torque from R. and L. cylinders $= (20 \times 10) + (20 \times 5)$
$= 300$ tons inches.

Each axle absorbs one-third of this $= 100$ tons inches.

The R.H. coupling rods are assumed to carry all the load in this position,

$\therefore$ Bearing load on leading and trailing crankpins $= \frac{100}{10}$
$= 10$ tons.

Bearing load on second crankpin $= 20$ tons.

For the Strength of the Coupling Rods.

With the L.H. crank on the dead centre,

Total torque from R.H. and centre cylinders
$= 20 \times \cdot867 \times 10 \times 2 = 346{\cdot}8$ tons inches.

If the leading wheels slip, the full torque of the centre cylinder is transmitted back through the R.H. leading coupling rod, giving a load on it of 20 tons.

If the total torque of 346·8 tons inches is absorbed equally by the second and trailing wheels, the trailing coupling rod will take a maximum load of $\frac{346{\cdot}8}{2 \times {\cdot}867 \times 10} = 20$ tons.

Alternatively, if the second pair of wheels slip, the total torque is absorbed equally by the leading and trailing wheels, and the trailing coupling rod still takes a load of 20 tons, while the leading coupling rod takes (apparently) no load at all. Hence the coupling rods would be designed to take 20 tons throughout.

The correctness of this treatment for bearing loads and stresses is confirmed by an examination of some existing successful designs.

Note that if the centre crank is on the dead centre, the two outside cylinders produce a total torque of 346·8 tons inches, as before, but in this position it cannot be clearly shown which coupling rods are transmitting the loads. Probably both sides are loaded for part of the revolution.

A six-coupled locomotive only has been examined here, but an eight-coupled locomotive can obviously be analysed on similar lines.

Four-Cylinder Locomotives.

Fig. 16a shows a diagram of a six-coupled four-cylinder locomotive, with all the cylinders driving the middle axle and with adjacent outside and inside cranks at 180°. Suppose the piston load = 14 tons for each cylinder and the stroke = 20″.

The maximum torque is applied when all the cranks are at 45° and all four cylinders can exert full piston load.

$$\text{Max. torque} = 14 \text{ tons} \times \frac{10}{\sqrt{2}} \times 4 = 396{\cdot}7 \text{ tons inches.}$$

This is absorbed equally by each pair of wheels, hence the coupling rods are transmitting $\frac{396{\cdot}7}{3} = 132{\cdot}2$ tons inches to the leading and trailing axles respectively.

With the cranks at 90° to each other, as in this case, the load is transmitted by the coupling rods on each side alternately through an angle of 90° (approximately), *i.e.*, the load is transferred from one side to the other four times in a revolution, and the transfer occurs at 45°, subject to slight variation due to clearances, unequal lengths of rods, etc. It follows that the coupling rods on one side only transmit the torque of 132·2 tons inches to the leading and trailing axles, giving a bearing load on the leading and trailing crankpins $= 132{\cdot}2 \times \frac{\sqrt{2}}{10} = 18{\cdot}7$ tons.

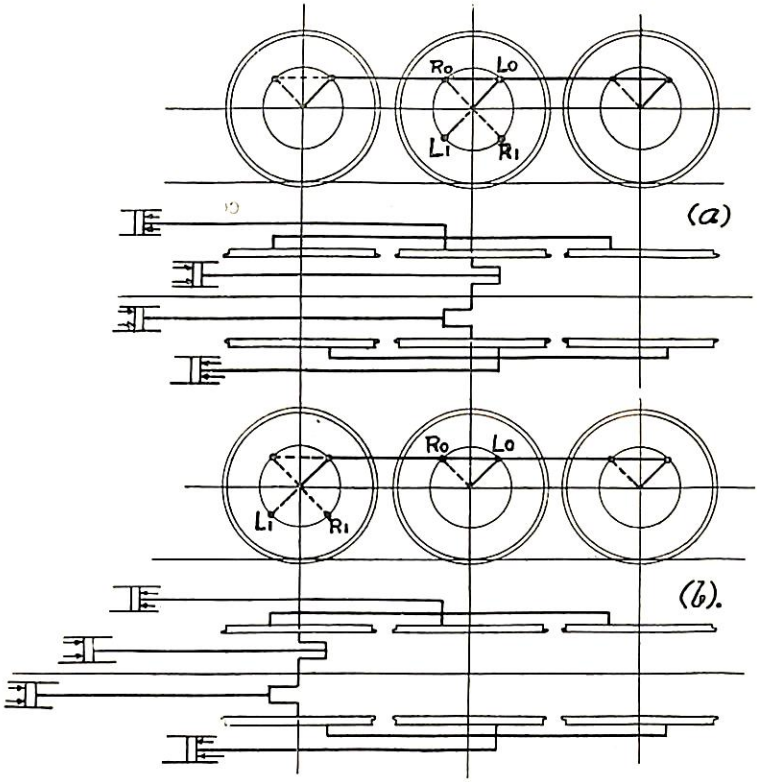

Fig. 16.

The bearing load on the coupling rod driving crank-pin is the sum of the other two, *i.e.*, 37·4 tons.

For the strength of the coupling rods, consider the driving wheels to be slipping, when the full torque of 396·7 tons inches is transmitted to the leading and trailing axles, giving a maximum load on each half of the coupling rod $= \dfrac{396{\cdot}7 \times \sqrt{2}}{2 \times 10} = 28$ tons.

If the inside cylinders drive the leading axle, as per Fig. 16b,

Max. torque applied to leading axle

$$= 14 \text{ tons} \times \frac{10}{\sqrt{2}} \times 2 = 198{\cdot}35 \text{ tons ins.}$$

Max. torque applied to intermediate axle

$$= 14 \text{ tons} \times \frac{10}{\sqrt{2}} \times 2 = 198{\cdot}35 \text{ ,,}$$

$$\text{Total (as before)} = 396{\cdot}7 \text{ ,,}$$

For the bearing loads, the trailing crankpin takes

$$\frac{396{\cdot}7 \times \sqrt{2}}{3 \times 10} = 18{\cdot}7 \text{ tons, as before.}$$

The leading crankpin transmits a surplus of

$$198{\cdot}35 - 132{\cdot}2 = 66{\cdot}15 \text{ tons inches,}$$

giving a bearing load of $66{\cdot}15 \times \frac{\sqrt{2}}{10} = 9{\cdot}35$ tons.

The coupling rod bearing load on the intermediate crankpin

$$= 18{\cdot}7 - 9{\cdot}35 = 9{\cdot}35 \text{ tons.}$$

For the strength of the coupling rods, consider the leading wheels to be slipping, when a torque of 198·35 tons inches is transmitted by the leading coupling rod, giving a maximum load

$$= 198{\cdot}35 \times \frac{\sqrt{2}}{10} = 28 \text{ tons.}$$

The trailing coupling rod would also take 28 tons.

Note that these two variations of a four-cylinder drive differ only in respect of the bearing load on the leading and intermediate crankpins. The leading crankpin would generally be made the same as the trailing one, unless space restrictions at the front end made it desirable to take advantage of the smaller bearing load. The bearing surface of the intermediate crankpin would be the greatest of the three, as it must be of large diameter to take the bending from the outside cylinder. Consequently the coupling rods for this split drive might well be duplicate with those designed for the case of all four cylinders driving the second axle.

It may be thought inconsistent to work out the loads for a three or four cylinder locomotive on the basis of maximum total torque, and at the same time to treat a two-cylinder locomotive as though one side were completely divorced from the other. This procedure for a two-cylinder locomotive is hallowed by long usage and the results are known to be satisfactory. With three or four cylinder locomotives (which, by comparison, are in a minority) the writer considers the maximum total torque to be the most rational basis of design. If bearing pressures and factors of safety as quoted for a two-cylinder locomotive are allowed, a satisfactory design is assured.

Crank Axle for a Three-Cylinder Locomotive.

CASE I.—*All cylinders driving the same axle.* Cranks spaced 120° apart. (See Fig. 16c).

If C = centres of axle journals in inches.
D = distance from centre of journal to centre of outside cylinder in inches.
R = radius of crank in inches.
P = piston load in tons.

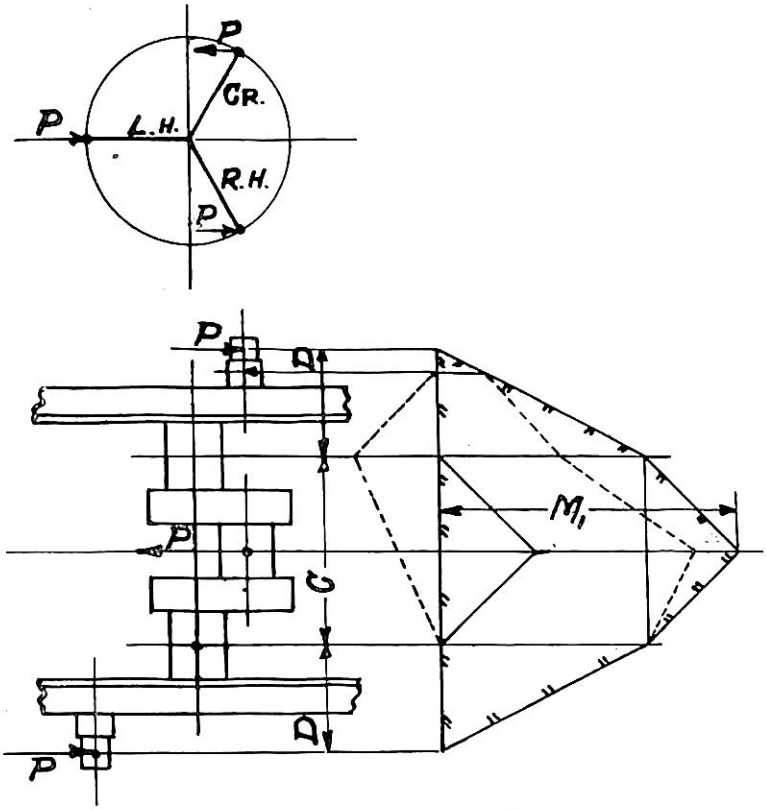

Fig. 16c.

The maximum horizontal bending moment occurs in the position shown in Fig. 16c, with the L.H. crank on the dead centre and all three pistons assumed to be taking full steam pressure, which represents the worst conditions of loading. The bending moment diagram shows the bending moments for the outside and inside cranks, together with the total bending moment. The maximum value occurs at the centre of the inside crank bearing.

Maximum horizontal bending moment,

$$M_1 = \left(\frac{P \times C}{4}\right) + (P \times D) \text{ tons inches.}$$

The twisting moment being transmitted through the centre crank bearing under the above conditions is small, and for the present purpose may be neglected. The vertical bending moment, due to the static load on the axle-box journals, may also be neglected, and the diameter of the centre crank bearing may be determined

by reference solely to the bending moment M_1, allowing a stress of 8 to 10 tons per square inch. An allowance of $\frac{1}{4}$" to $\frac{3}{8}$" for wear is then added to the diameter.

Note that the coupling rods are assumed to take no load. A considerable load *must* be taken by the coupling rod in the R.H. side in the position shown, as one pair of wheels cannot absorb the total torque produced by the loads on the R.H. and centre cranks. The dotted lines on the bending moment diagram (Fig. 16c) show that the effect of a load on the R.H. coupling rod is to reduce the bending moment on the axle, and hence reduce the stress in the centre crank bearing. However, the actual amount of such reduction cannot be accurately estimated.

At the centre of the axle-box journal, the maximum horizontal bending moment $M_2 = P \times D$ tons inches.

In combination with this, a twisting moment $T = P \times R$ should be allowed for.

Hence, maximum equivalent bending moment

$$= \tfrac{1}{2} M_2 + \tfrac{1}{2} \sqrt{M_2^2 + T^2}$$

Allowing a stress of not more than 9 tons per sq. inch, the diameter of the journal may then be determined. To this must be added the allowance of $\frac{1}{4}$" to $\frac{3}{8}$" for wear.

The diameter of the centre crank bearing will be equal to a slightly greater than the diameter of the axle-box journal. In practice they are often made equal, though at least one British standard class of locomotive has the crank $\frac{1}{2}$" larger than the journal.

CASE 2.—*Outside cylinders driving the second axle and inside cylinder driving the leading axle.*

The second axle in this arrangement is a straight axle, designed as for a two-cylinder locomotive, but allowing a maximum stress of about 10 tons per sq. inch.

For the leading cranked axle driven by the inside cylinder it is usual to make the journals the same as for the second axle. Frequently the centre crank bearing is the same diameter also. If so made, the crank axle is of ample strength. The strength of the centre bearing may be checked for bending due to the load from the centre cylinder, but the remaining loads on it cannot be accurately determined.

Crank Axle for a Four-Cylinder Locomotive.

Four-cylinder locomotive, with all cylinders driving the same axle. Adjacent inside and outside cranks at 180°. (See Fig. 16d).

If D = distance from centre of journal to centre of outside cylinder in inches.

E = distance from centre of journal to centre of inside cylinder in inches.

R = radius of crank in inches.

P = piston load in tons.

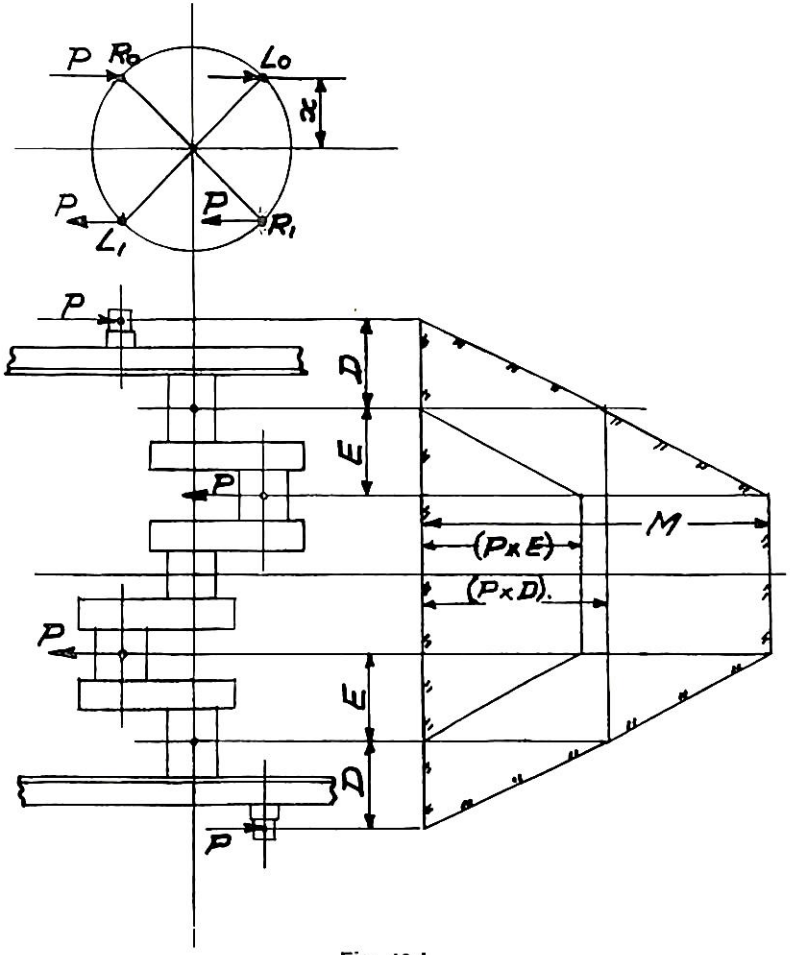

Fig. 16 d.

The maximum horizontal bending moment occurs when all the cranks are at 45°, as shown in Fig. 16d, and all pistons are taking full steam pressure.

The maximum horizontal bending moment $M = (P \times D) + (P \times E)$ and occurs in the centre portion of the axle, from centre to centre of the inside cranks.

The twisting moment T being transmitted through the body of the axle may be assumed to be that derived from two cylinders,

$$i.e., \quad T = 2P \times x = 2P \times \frac{R}{\sqrt{2}} \text{ tons inches.}$$

As before, the equivalent bending moment $= \frac{1}{2} M + \frac{1}{2} \sqrt{M^2 + T^2}$. A stress of not more than 9 tons per sq. inch should be allowed, and the usual allowance for wear added to the diameter.

The maximum horizontal bending moment in the axle-box journal occurs at the inner end of the journal, and may be scaled from the bending moment diagram. Combining this with the twisting moment T and allowing a stress of not more than 9 tons per square inch, enables the diameter of the journal to be determined, after which the usual allowance for wear is added.

As in the case of the three-cylinder design, no account is taken of the load on the coupling rod, as this cannot be accurately determined. It will have the effect of reducing the bending moment in a similar manner to that shown on the bending moment diagram for the three-cylinder locomotive (Fig. 16c).

If the drive is split, with the two outside cylinders driving the second axle and the two inside cylinders driving the leading crank axle, the second axle may be treated like a straight axle for a two-cylinder locomotive but allowing a maximum stress of about 10 tons per sq. inch. The diameters found for the journals of the second axle will be suitable also for the leading crank axle. The crank bearings on the leading axle should be checked for bending, the maximum bending moment in this case being $M = P \times E$ tons inches. This should be combined with a twisting moment $T = P \times R$, and a stress of not more than 9 tons per sq. inch may be allowed as before.

A *maximum* allowable stress for axles of three and four cylinder locomotives has been quoted in each case. Where weight and other conditions permit a lower stress is probably desirable, owing to the complex and uncertain loads.

Bogies and Trucks.

To permit a locomotive to negotiate curves freely, bogies or trucks are usually provided, except in the case of small or slow-speed locomotives. The term "Bogie" will be used here to indicate one having four wheels and "Truck" to one having two wheels. For passenger locomotives a leading four-wheeled bogie is most commonly used, but the weight to be carried is a deciding factor in this matter, and passenger locomotives with leading two-wheeled trucks are by no means uncommon.

The side-play of the bogie or truck is determined from the curve on which the wheel-base is laid down, being made just sufficient to allow the engine to pass comfortably round the minimum curve specified. Some form of side control gear is always provided to centralise the bogie or truck and to exert a guiding force on the main frames. Thus on entering a curve the bogie or truck is pushed off centre by the pressure of the rails on its wheel flanges, which brings the side control gear into operation and leads the driving

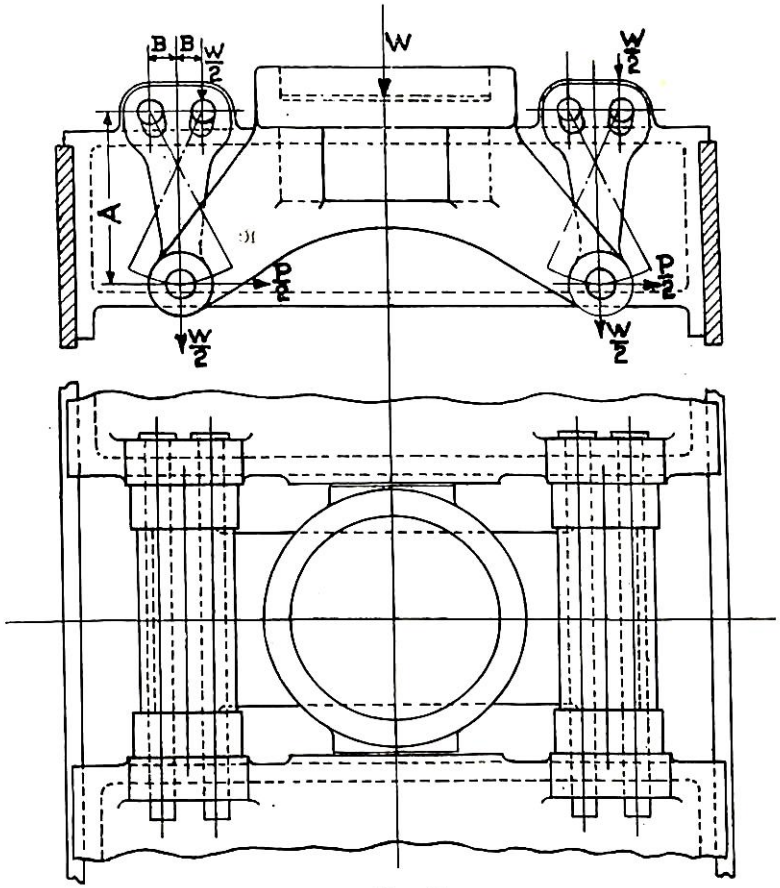

Fig. 17.

wheels into the curve. Without control gear there would be excessive wear on the leading coupled wheel flanges and rough riding at speed. On the straight track the side control gear tends to prevent "nosing."

Four-Wheeled Bogies.

The following notes cover the usual variations in design of modern four-wheeled bogies.

Swing Link Type is one in which the bottom pivot casting transfers the weight to the bottom pins of two pairs of triangular links, located transversely on either side of the centre. (See Fig. 17). Two pins pass through the top of each pair of swing links and transfer the weight to the bogie frame stay. In the central position both

top pins are under load, but as soon as the bogie is moved off centre the whole weight comes on one pin, producing a couple $= \frac{W}{2} \times B$, which is balanced by another couple $= \frac{P}{2} \times A$.

Hence the Initial Horizontal Control Load, $P = \frac{W \times B}{A}$, acting on the pivot casting as shown.

As the bogie moves further over, the horizontal distance B increases, and since P is directly proportional to B the control load increases in almost direct proportion to the side movement of the bogie. The dimensions A and B on the swing links must be arranged so that the initial value of P is sufficient and the final value is not excessive. Some remarks on the desirable loads are given later. This type of bogie was introduced many years ago and continues to be much in favour. It will be noted that the operation of the swing links lifts the front end of the locomotive slightly, causing an alteration in the weight distribution.

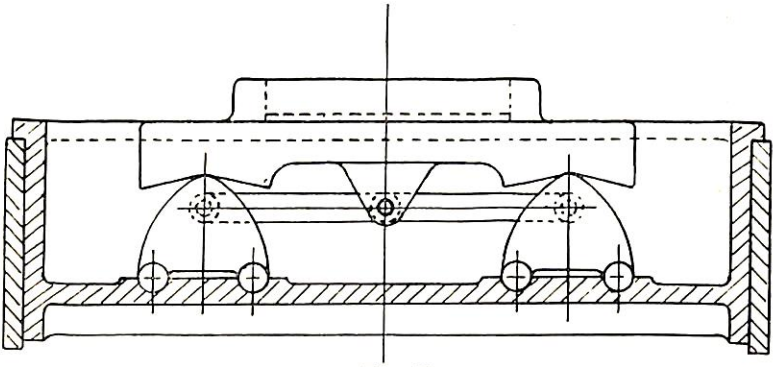

Fig. 18.

Swing links must be of ample strength and rigidity, as their duties are arduous. The shear stress in the pins is frequently not more than 1½ tons per square inch. There is, of course, some bending on the pins, since the links do not work in a jaw.

A variation of the swing link control is the constant load control, used in America to some extent. Fig. 18 shows this diagrammatically, from which it will be seen that the bottom pivot casting rests on the points of heart-shaped rockers, which in turn make contact with the bogie frame stay at two points. When the bogie is moved over, the whole weight comes on one point, as in the swing link, but

here the inclined faces on the pivot casting are so arranged that the point of contact of the rocker top moves as the bogie moves. The whole is arranged so that the side control load transmitted by links from the rockers to the pivot casting is constant. This type imparts a very decided lift to the front end of the locomotive.

Spring Side Control Type.

Side control by means of helical springs is now possibly the most favoured arrangement. The bogie bottom pivot in this case is a block sliding on faces on the bogie frame stretcher, and the springs may be located in various positions, of which several are indicated in Fig. 19. Two springs are provided, and in the best designs they are arranged so that both are compressed whenever the bogie moves off centre in either direction. This ensures that the controlling force is symmetrical for movement of the bogie to either side. An arrangement similar to (a) in Fig. 19, but with one spring only being compressed when the bogie moves over is also used. It is somewhat simpler, and appears to function well enough. However, experience has shown that if the two springs in such a bogie are not evenly matched the bogie has a bias towards one side, and if one of the springs breaks, the control is completely unbalanced. Where the springs are operated together, as in Fig. 19, the breaking of one spring leaves a weakened but still symmetrical control. Further, smaller springs can be used since the power of both is available.

The springs are put into position with an initial compression giving the desired initial control load. Since the final load must not be excessive, it follows that for a bogie with large side play one must design, for the space available, a spring having the lowest possible unit deflection (inches per ton), consistent with the required initial load, the working travel, and the maximum safe stress in the spring.

If a constant control load is desired, Fig. 19 (c) may be altered, as shown by (d). Here the rocker arms carry rollers which bear on the centre casting, and the set of the arm carrying the roller is so arranged that when the bogie centre moves over the increase in the effective length of this lever from its fulcrum compensates for the increase in the load on the spring as it is compressed. Hence a constant load on the roller can be obtained.

Since the helical springs have to be initially compressed and then got into position, it is important that one should be quite clear about how this is to be done, and make the best possible arrangements for it. Wherever possible, the bogie design should permit a new spring to be inserted without lifting the locomotive and running the bogie out. A suitable provision for inserting a spring is to form the end bearing washers with a lug top and bottom and compress the spring by means of screwed bars passed through holes in these lugs, the bars being removed after the spring is in position. In the type shown in Fig. 19 (a) (based on Southern Railway prac-

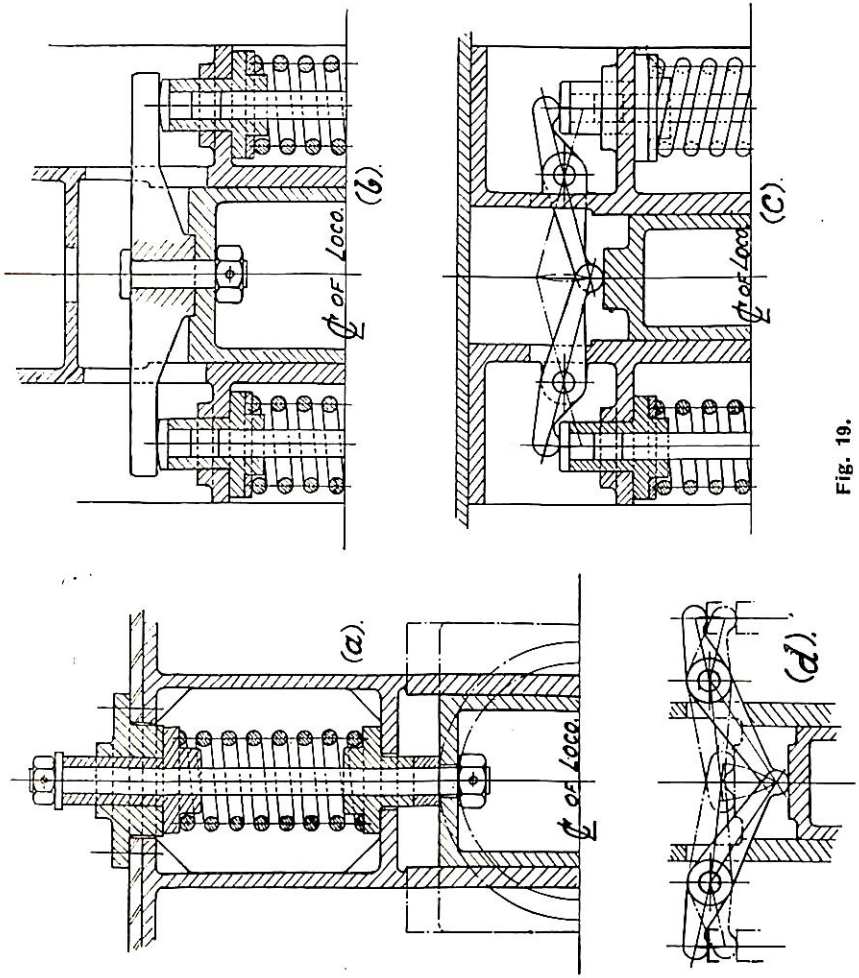

Fig. 19.

tice), the springs are inserted through the bogie frame and compressed by a cover flange on the outside.

A comparison of the merits of swing link and spring-controlled bogies shows an advantage to the latter on the following points :

The sliding centre presents a much larger bearing surface to take the weight than the pins of the swing link type.

The friction of this surface helps to damp out oscillations of the bogie.

Movement of the bogie does not affect the weight distribution of the locomotive, as there is no lifting effect such as already mentioned in connection with swing links.

Where a bogie takes the weight on two side bearers instead of the centre pivot, swing links cannot be applied.

An alternative to the use of helical springs is a pair of laminated springs coupled together by adjustable links at their ends. They are located in the bogie frame stretcher by suitable stops, and are deflected by movement of the bogie centre in either direction. Fig. 20 illustrates this type.

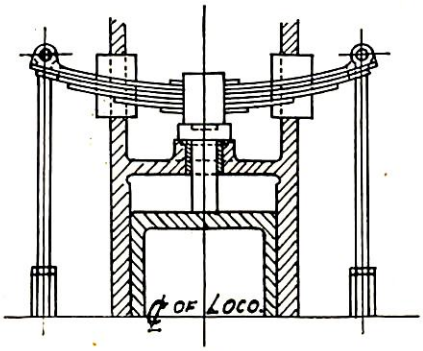

Fig. 20.

The side control loads to be allowed are difficult to fix from theory, since they are bound up with the behaviour of the locomotive as a whole on curves. For those who are mathematically inclined, a paper on this subject by the late S. R. M. Porter on "The Mechanics of a Locomotive on Curved Track" was published in the *Railway Engineer* of July, 1934, and onwards. This expounds different theories on the subject very fully. From an examination of existing practice it would appear that the side control loads may vary considerably without materially affecting the running of the locomotive. In general, a very small control force may lead to oscillations and to excessive wear on the flanges of the leading coupled wheels, while an excessively large control force tends to derail the bogie. One rough method of estimating the load required is as follows (see Fig. 21).

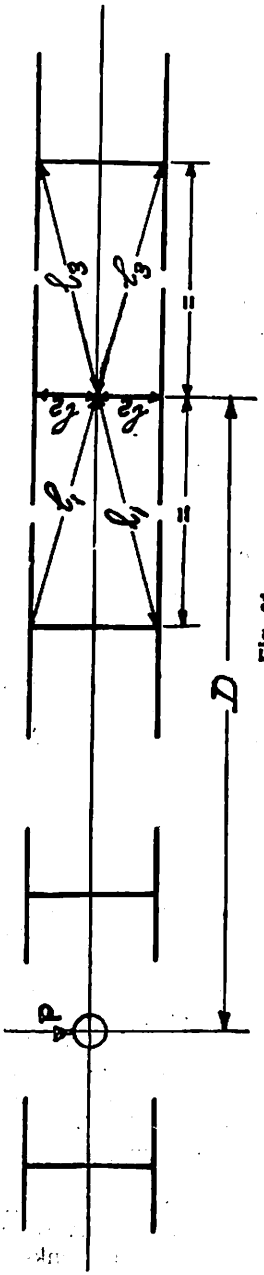

Fig. 21.

When the locomotive enters a curve the bogie is displaced first, and through the control force swings the main frames round to follow. If it is assumed that the locomotive pivots about the centre of gravity of the coupled wheel-base; that this causes the coupled wheels to slide by a small amount on the rails; that the coefficient of sliding friction between wheel and rail = ·25 = μ.

Then if L = rail load on each pair of coupled wheels.
$l_1\ l_2\ l_3$ = distances from assumed pivot point to contact point on rail.
D = distance from assumed pivot point to bogie centre.
W = load on bogie centre.
P = required side control force exerted by springs.

$$P \times D = \cdot 25 \{ (L \times l_1) + (L \times l_2) + (L \times l_3) \}$$

The friction of the slide is here neglected. It assists the springs when entering the curve and resists them when leaving it.

The maximum value of the control load of the springs is approximately equal to the value of P thus found for a locomotive with a leading bogie only.

Where a trailing bogie or truck is fitted, also with side control gear, the value of P found might be about equal to the *average* value of the load on the springs, since the trailing bogie acts to some extent in opposition to the leading bogie when on a curve.

The method just propounded is to be regarded as a suggested basis of comparison. The writer is not prepared to assert that it represents the true facts.

On another basis, the initial side control load is frequently ·08 to ·11 of the pivot load. Probably this basis is accurate enough for most cases, though as a general statement it is lacking in logic.

It is pointed out by Porter that a locomotive designed to run always forward, and having both leading and trailing bogies or trucks, requires at the trailing end only sufficient control force to overcome friction of slides, etc., as the leading end only should guide. For a locomotive designed to run both ways, both must be capable of guiding. As already stated, they act in opposition, but the leading bogie is displaced further than the trailing one, and hence exerts the greater load.

It is important to note that a locomotive with bogies or trucks at both ends should not have swing link side control on both. If made so, the coupled wheels would be relieved of load to a dangerous extent due to the lifting action of the swing links when traversing a curve, which is just the time when full adhesive weight is most needed for traction purposes and for safe riding.

In a locomotive of symmetrical design such as a 4 - 6 - 4, it is undesirable to make the bogies duplicate, as the locomotive may tend to sway badly, due to possible synchronising of oscillations in the control springs of the two bogies. A common procedure in such cases is to make one bogie with swing links and the other with

spring control, or one with helical springs and the other laminated springs.

The load is applied to bogies in three ways :—

By a spherical pivot.
By a flat pivot.
By two side bearers.

The spherical pivot is used to some extent on the Continent. The flat pivot is the most common arrangement, but where the track is of a high order of perfection, side-bearers are used to some extent. With side-bearers, the weight is transferred directly from the main frame to the bogie frame, hence the pivot centre and bogie frame stretcher can be made lighter. The centre and slide function only to propel the bogie and operate the control gear. The bearing pressure on a flat pivot is generally about 100 lbs. per sq. inch, and efficient lubrication is provided.

The bogie frame stretchers are usually of cast steel, though it is now possible to fabricate such parts from steel plates by electric welding. So far, welding has not been much used on locomotive bogies.

Bogie frames are most commonly of steel plate of the same thickness as the main frames. The bogie frame stretcher should be riveted to the frames with cold turned rivets, and the flange lipped over the edge of the frame if heavily loaded. Where the bogie has four independent bearing springs it is usual to arrange them inside the bogie frames, while for compensated arrangements the springs and gear are outside the bogie frames.

An alternative to the plate frame for bogies is a stout rectangular bar forming the main top member, with the axlebox guides, etc., bolted to its underside, and a tie-bar joining the bottom of the guides.

Bogie Bearing Springs.

Laminated bearing springs appear to be the most satisfactory. They have a deflection per ton of $\frac{3}{16}$" to $\frac{1}{4}$", and follow the design of the coupled springs. If the coupled springs are compensated, the bogie springs are usually compensated also. This is generally achieved by fitting one large inverted spring on each side (see Fig. 22 a). The buckle is pivoted on a bracket on the outside of the bogie frame, and the load applied to the ends by links from a beam whose ends rest on the front and back axle-boxes. Adjustment is provided for in the spring links or in the bearing pieces which rest on the axle-boxes. To reduce the shock on the laminated spring, auxiliary springs are sometimes fitted on each spring link. These may be small helical springs or rubber pads reinforced with steel.

The spring beam may have a maximum bending stress of about 6 tons per sq. inch, while the spring links have usually a tensile stress of about $2\frac{1}{2}$ tons per sq. inch. Note that when stressing the

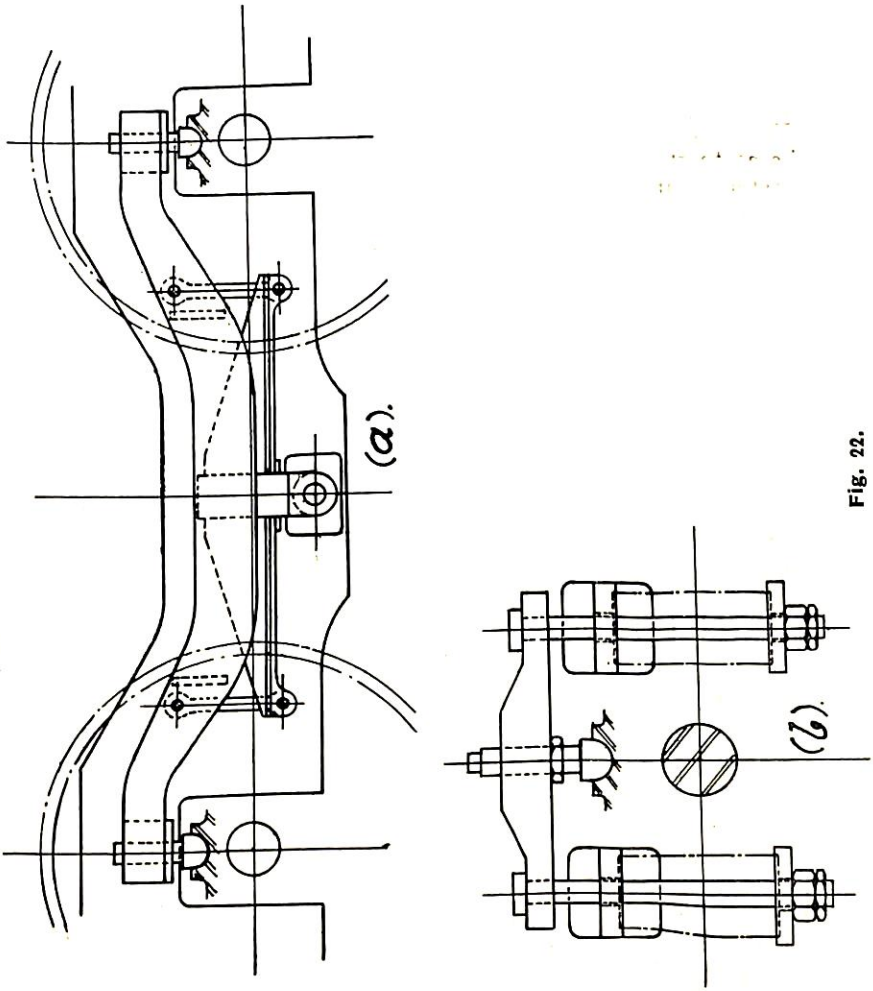

Fig. 22.

bogie frame stretcher for bending the support points of the "beam" are the brackets taking the spring buckles, and *not* the frame plates. Also, it is a simply supported beam, and not one with ends fixed.

Where helical bearing springs are used, a compact arrangement is obtained by fitting one on each side of the axle-box and by loading the pair by means of a short beam bearing on the top of the axle-box. (See Fig. 22 b). Here again a convenient adjustment can be arranged for in the bearing piece on the axle-box.

Bogie Axles.

The bearing pressure allowed on bogie axle journals is usually about 180 to 200 lbs. per sq. inch, based on the projected area of the journal. The arc of contact of the bush may be reduced to about 80°, provided the bogie wheels are not braked. Hence the actual bearing pressure is higher than stated. The figures given for projected area allow for this. It is desirable that the axle-box face which bears on the wheel boss should be lined with white metal, as the bogie side control forces cause a considerable thrust on these faces, and they should also be efficiently lubricated.

For the diameter of a bogie axle, if the bending moment due to the bearing load only is considered, *i.e.*, if it is taken as a beam supported at the centres of the rails and loaded at the centres of bearings, a stress of 2 to 3 tons per sq. inch should be allowed for the body of the axle, the journals being made $\frac{1}{4}$" or $\frac{3}{8}$" larger in diameter than the body to allow for wear. Going a little deeper into the actual loads on the axle, the latter must withstand a load between rail and wheel flange due to the guiding function of the bogie, which will be something in the nature of a shock load. Also, the bearing load on the journals may vary due to rolling of the locomotive, and to the lifting action of swing links, if fitted on the bogie.

Considering only the bearing load and the thrust between wheel and rail, a value for the latter of 40% of the rail load per axle has been suggested. This corresponds roughly to taking half the maximum control load as acting on one wheel, and doubling it for shock load.

To illustrate this, consider a 4'-8$\frac{1}{2}$" gauge bogie with 3'-6" wheels and rail load 12 tons. Bearing load 5$\frac{1}{2}$ tons per journal. Centres of bearings 3'-8". Centres of rails 4'-10$\frac{1}{2}$". Side control gear with maximum load of 5 tons.

Bending Moment due to Bearing Load = 5·5 tons × 7$\frac{1}{4}$
= 39·9 inch tons.

40% of 12 tons = 4·8 tons side thrust at rail.

Bending Moment due to couple produced by this load
= 4·8 tons × radius of wheel
= 4·8 × 21 = 100·1 inch tons.

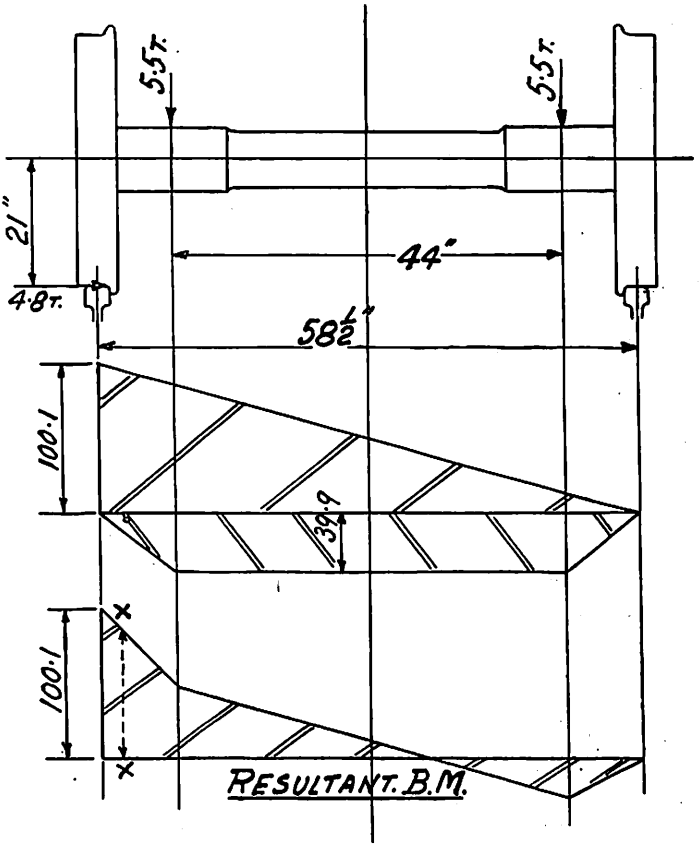

Fig. 23.

Reference to Fig. 23 shows that for an axle with the bearings inside the wheels, as shown, these two bending moments are opposed to each other, so that the maximum resultant occurs in the L.H. wheel seat and is that due to the couple only.

(If the bearings were outside the wheels, the maximum resultant bending moment would be the sum of the two and not the difference).

In the example shown, it would be reasonable to assume that this maximum bending moment acts at the face of the wheel boss, and the section resisting bending would be the journal diameter, less about $\frac{1}{4}$" or $\frac{3}{8}$" for wear. The centre part of the axle is generally about equal in diameter to this lower limit of journal diameter.

If a stress of 2·5 tons per square inch is allowed for bending due to the bearing loads only, it follows that, in the example, the actual stress would be at least $2{\cdot}5 \times \frac{100{\cdot}1}{39{\cdot}9} = 6{\cdot}27$ tons per sq.

inch, which is quite reasonable as an axle stress and would still allow for increase due to variation in the bearing loads.

Note that designing the axle for the stress given on the basis of bearing loads only is really a basis of comparison, justifiable on the assumption that wheel diameters, etc., do not vary much. The writer suggests checking it both ways.

The relative movement between a bogie and the main frame should be carefully tried out for clearances at all points. (This is mentioned only as a warning to the unwary). Also, since the bogie wheel often moves right under the main frame, a working clearance must still remain after allowing for the following :—

Wear of bogie centre liner.
Wear of axle-box brasses.
Wear of axle journal.
Axle-box raised to its stop.

Stops are fitted on the bogie to limit the travel of the bogie centre, and on the main frames to prevent the bogie from swinging right round and causing damage to other parts in the event of a derailment.

(Brakes for bogies are discussed under Brake Gear).

Two-Wheeled Trucks.

The following are the most common types of two-wheeled trucks now in use :—

Radial arm truck with swing link side control.
Radial arm truck with helical spring side control.
Radial arm truck with laminated spring side control.

A Swing Link Truck takes the weight on a centre bearer. Compensation between the truck and the leading coupled springs is arranged by making the top centre bearer slide vertically in a guide in the main frame, the sliding portion being a hollow pillar of large diameter. Inside this pillar is a link connecting to the end of a compensating beam, which is taken back on the centre line of the engine and connected to the leading springs by a cross beam.

In one design (see Fig. 24), the two axle-box bearings are incorporated in one casting, and two laminated bearing springs are placed transversely on the bogie, one in front of, and one behind, the axle-box casting. Each end of these springs is carried in the bottom of a three-pin swing link. A large pin passes through the buckles of both springs, and the truck centre bearer projects down in the form of a saddle piece to engage this pin. The swing links are connected at the bottom by links to the centre casting, while their top pins pass through the top of the axle-box casting. The radial arm, usually built up from steel bars or angles, is bolted to the axle-box casting and pivoted to a pin on one of the frame stays.

Where the truck is independently sprung, the construction could be as described, but with the top centre bearer rigidly attached to

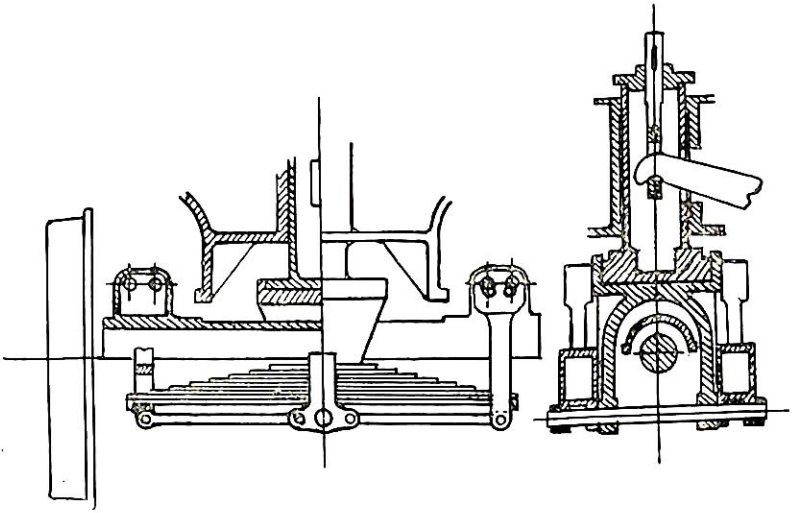

Fig. 24.

the main frame stay. Another form of independently sprung truck (see Fig. 25) has normal axle-boxes working in guides in small plate frames. The frames are stayed together and attached to the radial arm. A laminated spring may be fitted over each axle-box with links to the truck frames, or a pair of helical springs loaded by a bridle over the axle-box may be fitted, similar to those mentioned for four-wheeled bogies. The side control in this case consists of links at each side, whose lower ends are on the bottom centre bearer casting, whilst the upper ends are on the truck frame stays. When the truck is central, both links are inclined outwards. It is argued that when moved over on a curve this has an advantage in allowing the outer wheel to take the greater weight; while the inner wheel slips slightly due to the shorter rail.

A similar pair of links connects the top centre to the main frame stay, being inclined the opposite way so that the bogie can move over without causing the pivot to bear on one edge only.

Trucks with Spring Side Control Gear.

A common example of this type (see Fig. 26) has the two axle-box bearings embodied in one steel casting, to which the radial arm is bolted. The bearing springs are on the main frame, and bear on the top of the axle-box casting at each side through a spring pillar and a sliding block. This block slides on a renewable liner, and works in an oil bath. The side control spring is mounted on a bracket from the main frame stretcher in front of the axle-box casting. A spindle and sliding bush project at each end of the spring to engage lugs on the axle-box casting, whereby the spring is compressed when the truck moves off centre in either direction.

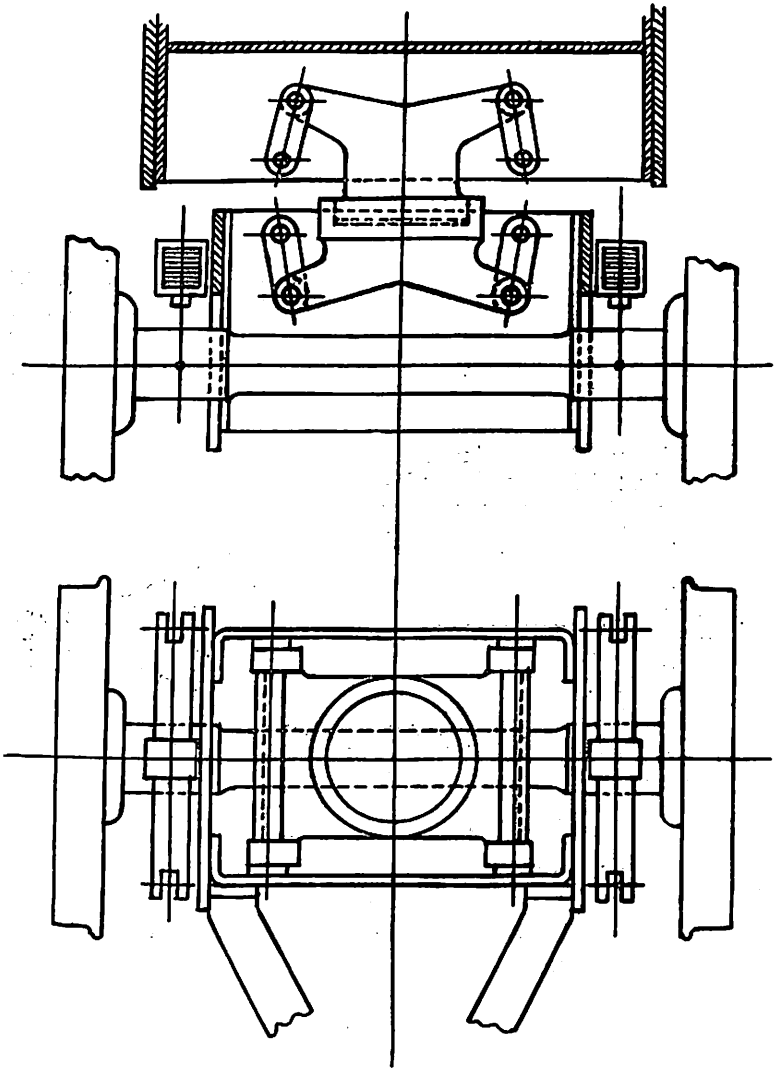

Fig. 25.

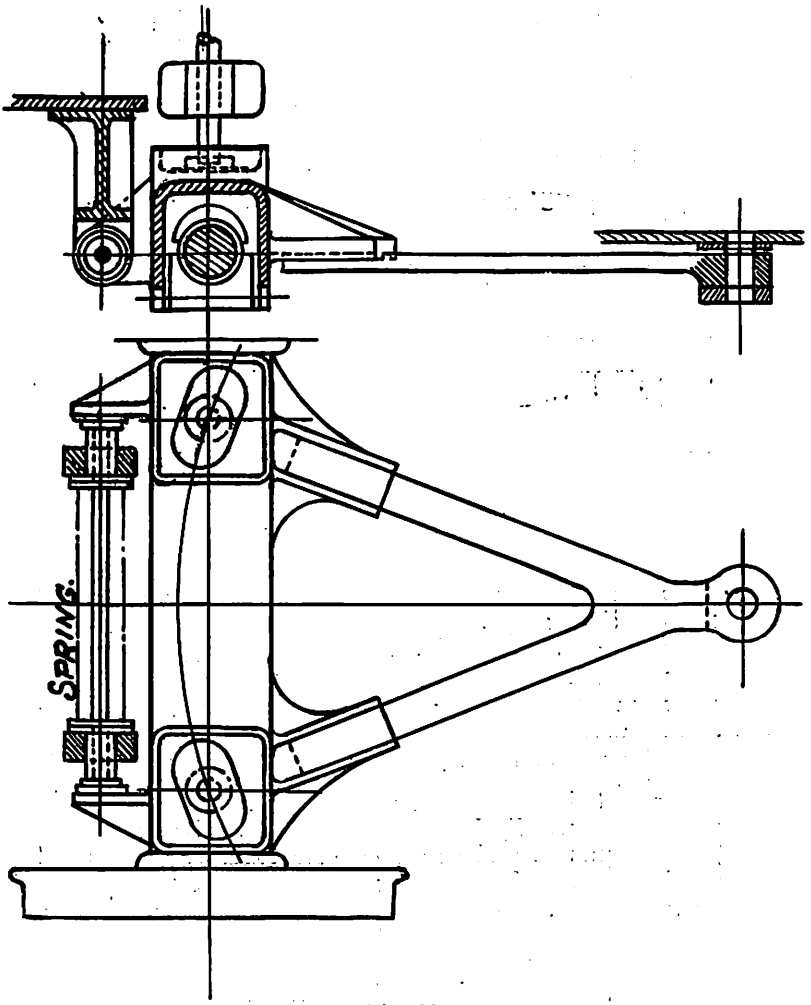

Fig. 26.

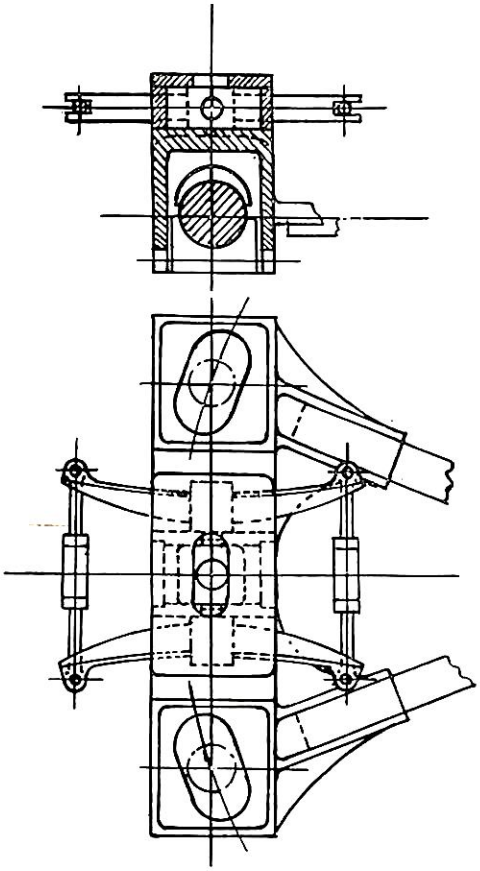

Fig. 27.

When laminated side control springs are fitted to a truck of this type they are usually located above the axle-box casting (see Fig. 27). The pair of springs are coupled together by adjustable links, and both are deflected when the truck moves off centre, through the buckle of one spring engaging a projection on the main frame stay.

Trailing Two-wheeled Trucks assume a different form, due to being located under the firebox. Such trucks generally have the axle-boxes and springs mounted outside the wheels, on a plate frame bolted to the radial arm. The axle-boxes may be bolted rigidly to the truck frame, with the bearing springs on the main frame, bearing on the axle-box through a spring pillar and sliding pad, or the axle-boxes may work in guides, with the bearing springs carried on the truck frame. In the latter arrangement (see Fig. 28)

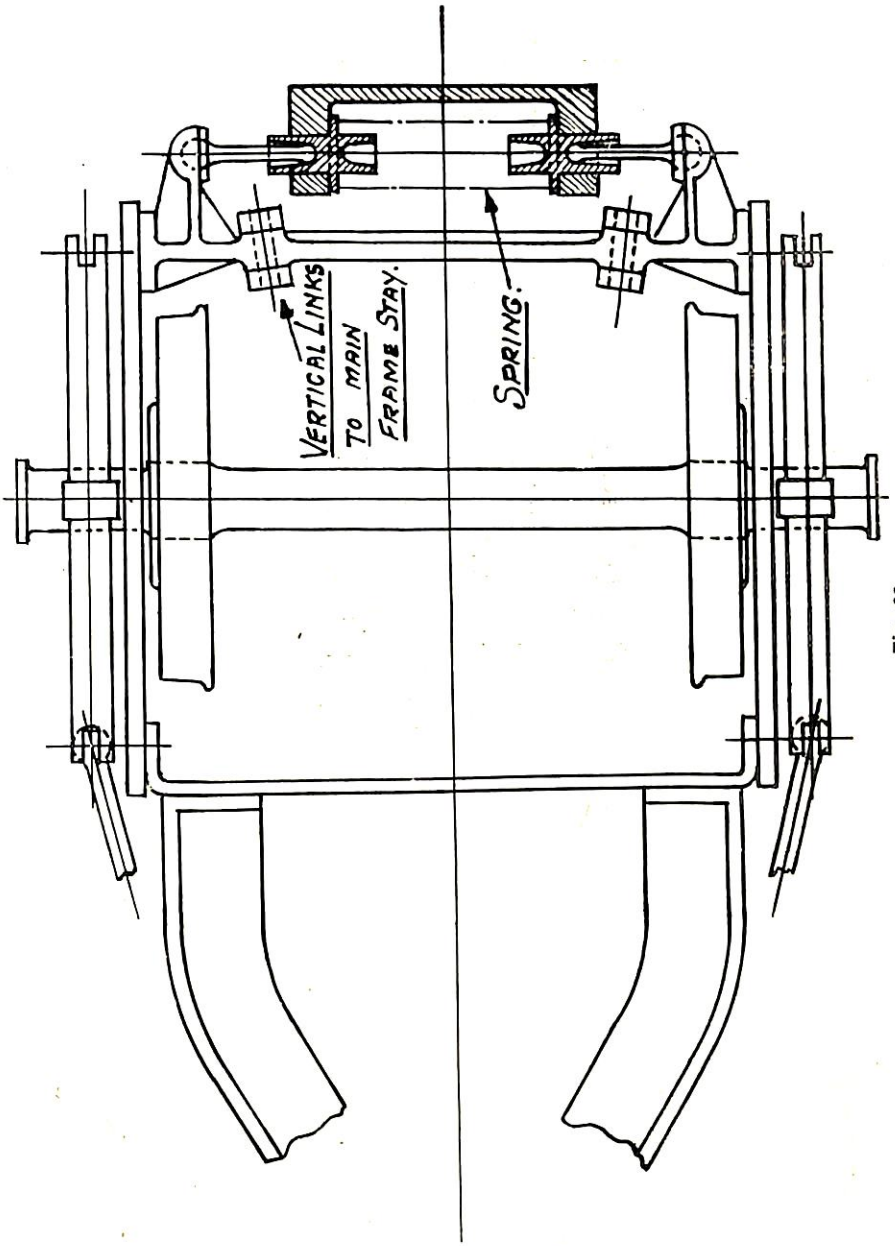

Fig. 28.

the weight is applied to the truck frame at the back end through swing links. Such swing links should be as long as possible, and their pins should point towards the truck pivot point. Ample clearance is necessary in the pin holes because of the radial movement of the truck frame. The truck bearing springs may be connected through compensating beams to the trailing coupled springs. Where the spring is on the truck frame some form of knuckle joint or spherical end is necessary on the spring link, because of the movement of the truck spring relative to the compensating beam. Side control springs (usually a nest of two) are carried from the main frame stay behind the truck frame, and are usually compressed by a pair of ball-ended struts secured in sockets on the truck frame. When designing the spring control for the type with the swing links, it must be remembered that the swing links exert a considerable centring force. Where a large side-play is required (say 6″ or more), this type of truck is not altogether satisfactory, as the total control force may become excessive and the lifting of the back end due to the swing links may be too great. Roller side bearers may be used instead of the swing links.

The remarks already made regarding bogie axle stresses and bearing pressures apply also to truck axles, and here again it is desirable to line the axle-box faces with white metal and attend to their lubrication in order to deal with the thrust on the wheel boss.

To fix the length of radial arm for a truck, Baldry's Rule is generally applied (see Fig. 29).

If Y = required length of radial arm.
T = distance from centre of rigid wheelbase to centre of truck axle.
F = half the rigid wheelbase.

then $Y = \frac{1}{2}.\left(T - \frac{F^2}{T}\right)$

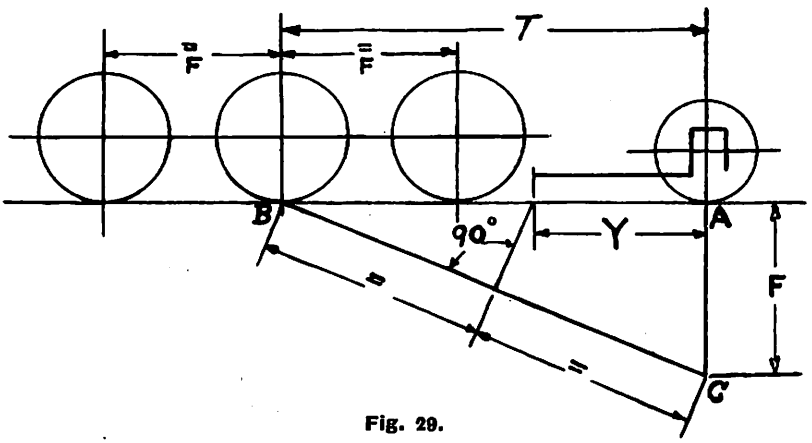

Fig. 29.

or graphically set down AB = T, and AC = F. Bisect BC and draw a line at right-angles to intersect AB. This gives the required length Y.

This rule is mathematically correct on the assumption that the leading coupled, trailing coupled, and truck wheels all have equal clearances between flanges and rail. It is sufficiently accurate for most cases, and in any case a few inches variation from the length obtained will not affect the running materially.

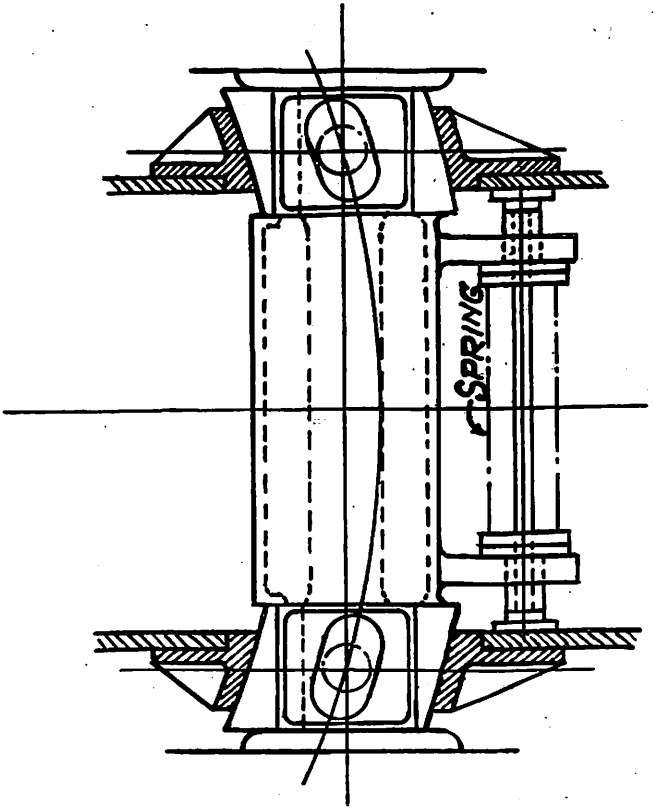

Fig. 30.

Radial Axle-boxes.

Although not really forming a truck, radial axle-boxes have exactly the same function to perform. The axle-boxes work in curved guides in the frame. The radius determined by Baldry's Rule is generally too short for application here (for constructional reasons) and a suitable compromise is made. Side control, where required, is effected either by a spring arrangement or by the "Cartazzi" type sliding top on the axle-boxes. The older form is

of the spring-controlled type (see Fig. 30) having the two axle-boxes formed in one casting on which the radial faces are made. The control springs are mounted above, below, or alongside the axle-box casting, suitable lugs being provided to engage the spring. Either helical or laminated springs are used. The bearing springs have a pillar and a sliding block bearing on the axle-box.

Such axle-boxes are of rather costly construction, and the working parts are all between the frames, where they are sometimes not very accessible.

The "Cartazzi" sliding top radial axle-box is now frequently fitted. Here separate axle-boxes are used, each working in a pair of curved guides on the main frame, outside the wheels. The "Cartazzi" top consists of a heavy steel block which is constrained to rise vertically in the guides. The underside of the block and the top of the axle-box are machined with inclined faces as shown in Fig. 31, so that when the axle-box moves to one side the block slides up one of the inclined faces. The spring load on the top tends to return the axle-box to its normal position. The incline is usually about 1 in 6 to 1 in 8.

As an example, if the spring load is 8 tons and the incline 1 in 8, the restoring force is the horizontal component of the load on the spring (neglecting friction),

i.e., Restoring force = $\frac{8}{8}$ = 1 ton per axle-box.
= 2 tons per axle.

Such axle-boxes should have a generous length of journal relative to the diameter (about 2D), since the loading is eccentric when the axle-box moves over. For a very large side-play the type is unsuitable. Good lubrication of the inclined faces and the face bearing on the wheel boss is essential.

Where only a small side-play is required (say less than 1″ each way), "Cartazzi" axle-boxes may be used, mounted in straight axle-box guides. With such an arrangement the wheel does not present itself to the curve in the correct way, but nevertheless it is known to work quite satisfactorily within that range of side-play. Thrust collars are necessary on the axle, and the journal should be of generous length in relation to its diameter (about 2D), because of the load moving off centre. As with all sliding top axle-boxes, lubrication is important on the sliding and thrust faces.

Axle-boxes.

Locomotive axle-boxes seem to be capable of an infinite variation in detail, which would require many pages of illustrations to cover. Briefly, the main variations are as follows :—

Axle-boxes of solid gunmetal or bronze.—These are largely used on some railways for coupled axles. The initial cost is heavy, but they are re-melted when scrapped. A strong point in favour of solid bronze is its ability to dissipate heat quickly.

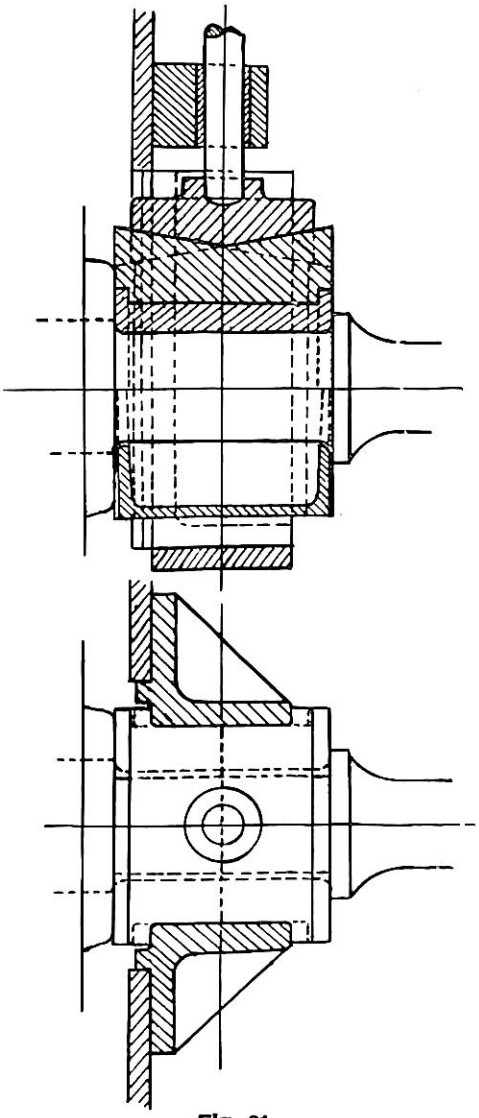

Fig. 31.

Cast-Iron Axle-boxes.—These are only used on small locomotives. Gunmetal bushes are fitted.

Cast Steel Axle-boxes.—These are now used extensively for all axles. The gunmetal bushes may have a flat top, bearing on a face in the top of the axle-box and located by a spigot, or (more frequently) they may be of half-round section pressed into the axle-box and located by screws. The bushes generally have white metal inserts, which may be in the form of rectangular pads let into the bush, or strips running the full length. In the latter case the recess in the bush has a serrated face cast or machined to take the white metal. Sometimes the face bearing on the wheel boss is serrated and lined all over with white metal, or alternatively white metal studs may be inserted in this face of the axle-box. On some railways a renewable liner is fitted on the face of the wheel boss. This liner is of cast iron or gunmetal, made in halves, and it is fixed by screws to the wheel.

The axle-box faces bearing on the guides may have renewable gunmetal liners screwed to the axle-box, or they may be lined with white metal. The flanges engaging the guides have a normal clearance of $\frac{1}{16}''$, frequently made parallel for about 3″ at the centre and then inclined to about 1 in 60 to give clearance in the event of rolling of the locomotive.

Taking the normal load on the guide face as $\frac{P}{n}$,

where P = piston load
and n = number of coupled axles.

The bearing pressure on this face is (very approximately) about 200 lbs. per sq. inch.

Lubrication.—If oil is fed from the top, the best practice is to lead it to two grooves, placed each about 40° from the vertical centre line. The grooves should be of generous depth, preferably cut in the gunmetal part of the bush rather than in the white metal. The axle-box keep forms an oil reservoir, and contains a pad which is pressed up against the journal by light springs, thus oiling the surface continuously. The bush should be backed off slightly for about 2″ above the horizontal centre line, forming a wedge-shaped clearance tapering from 0 to about ·006″. This helps to maintain a film of oil between the bush and the journal.

Grease lubrication for coupled axle-boxes is used on some railways. A perforated brass strip envelops the lower half of the journal, and a block of grease is pressed up against it by springs. A dependent chain or wire gives visible indication of the state of the supply. The bushes are not lined with white metal, as the working temperature is high for the grease to melt, and would be in danger of melting the white metal also.

For either grease or oil, it is now considered good practice to design the keeps so that they can be pulled out easily between the frames for inspection or refilling. Previously, keeps could not be removed without taking the horn-clips off, or even dropping the wheels.

Dust shields of leather or hemp rope are sometimes fitted on the inner face of the axle-box.

It is not now usual to have collars on the axle at the inner end of the journals, unless required for side-play or other reasons.

Axle-box Guides.

Axle-box guides are generally made of cast steel. Cast iron may be used for small locomotives, provided they are thick and well ribbed. They are secured to the frame by driven bolts or cold turned rivets. A parallel liner may be bolted to one face and a tapered wedge to the other, the latter to take up wear. Alternatively, parallel guides may be provided and liners on the axle-box renewed when necessary. The wedges have an incline of about 1 in 10, and are capable of being raised about 2" by means of a screw adjuster at the horn-clip. On some Colonial railways an automatic device is used (the "Franklin" wedge), consisting of a spring arrangement on the horn-clip which automatically adjusts the wedge.

The wedges are sometimes towards the cylinders and sometimes away from the cylinders—it does not appear to matter much.

When underhung springs are used (see Fig. 32 a), the two guides are conveniently cast in one piece, the part connecting the tops providing a useful reinforcement to the frame over the gap for the axle-boxes. In this form they are generally known as horn-blocks. The top corners should be thickened up and well-ribbed, and the bolts to the frame should be kept back from the corners of the gap in the frame plate.

The bottom of the guides are joined by a horn-clip of cast steel or forged steel. It is lipped over the guides, and sometimes grips the frame also. Alternatively it may grip the frame only. Horn-clips should be strong and rigid, the section being frequently determined from the stress set up when the locomotive is jacked or lifted from the ends.

On bar frame locomotives the frames are central with the bearings, and the liner and wedge fit directly on the frame, while the horn-clip grips projecting lugs on the bottom edge of the frame. (See Fig. 32b).

Springs and Spring Gear.

Bearing Springs.

Locomotive bearing springs are now usually of the laminated type. Helical springs have been extensively used in the past, but are now uncommon on new designs, except on bogies. The laminated

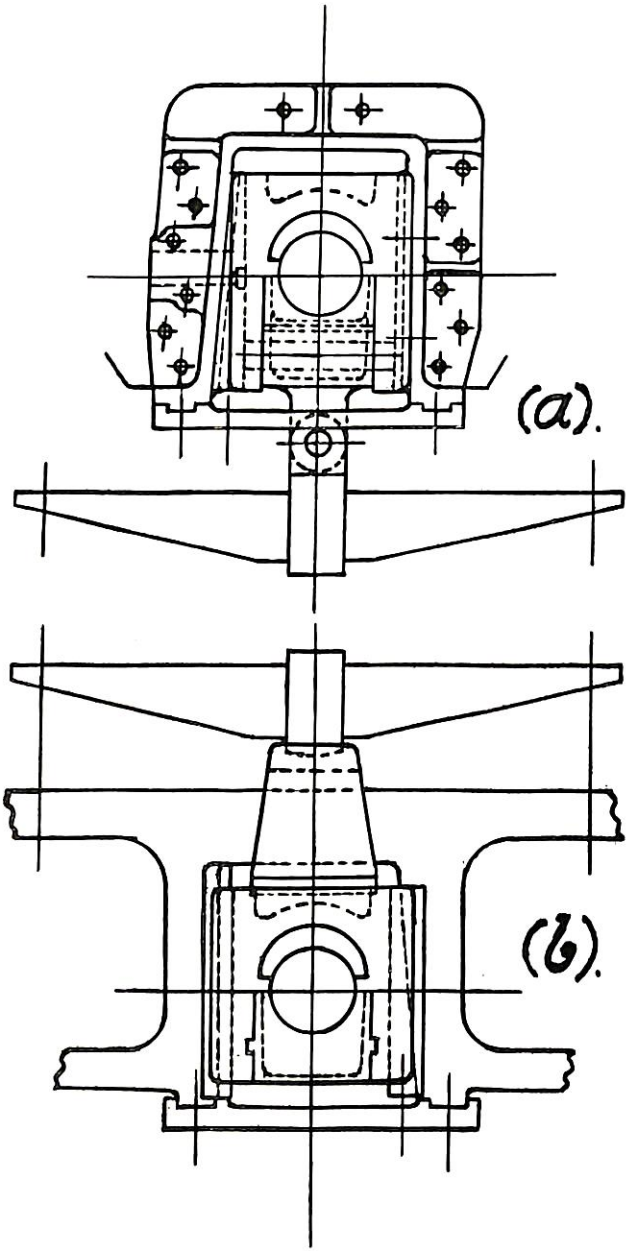

Fig. 32.

spring, although heavier and more expensive, has better characteristic riding qualities than the helical spring, and is less liable to cause a serious breakdown. A broken helical spring requires immediate replacement, whereas a laminated spring will continue to function even with several of its plates cracked. Also, of course, it can be readily repaired by inserting new plates.

Springs should have a unit deflection, $\delta = \frac{3}{16}''$ to $\frac{1}{4}''$ per ton of load, and the plate stress at normal load may be 50,000 to 60,000 lbs. per sq. inch for springs of carbon steel. A spring of generous span, with a moderate number of plates of suitable thickness is preferable to one of very short span having a large number of thin plates. The wheel diameter often limits the possible length of springs, but generally speaking, for coupled wheels the span is seldom less than 2′-6″ and may be as much as 4′-0″.

The question of whether coupled springs should be "underhung" or "overhead" appears to be rather debatable. The "overhead" arrangement is best for the spring itself, as the sides of the buckle are not then called upon to transmit the load to the spring, as in the case of the underhung spring. Also the "overhead" spring is not liable to the same extent to corrosion from wet and dirt.

Some of the factors controlling the position of the springs are as follows :—

If the coupled wheels are of small diameter (say less than 3′-6″), it is difficult to obtain sufficient rail clearance with underhung springs, and if compensating beams are provided it is necessary to connect to the springs by short compression links, which are slightly unstable under load. For a standard gauge locomotive with such small wheels, it is therefore preferable to put the springs overhead, provided they are accessible for removal in that position. On the other hand, for a narrow-gauge locomotive the transverse centres of springs should be as great as possible, and for a plate frame locomotive this is best achieved by using underhung springs, if rail clearance will permit. On Bar Frame locomotives the overhead spring is most convenient, being located directly above the frame and accessible from the outside.

For Plate Frame locomotives where it is possible to use underhung springs with spring links in tension, an advantage lies in the fact that axle-box guides of the horn-block type, with the bridge piece across the top can be conveniently used. Also, the spring brackets and axle-box guides may have less overhang from the frame than would be the case with overhead springs.

The normal load on the springs is the rail load less deadweights. the deadweights comprising wheels and axle, axle-boxes, parts of coupling rods and connecting rods, etc., and the springs themselves.

The British Standard Test Deflection for laminated springs (See B.S. Specification 24/6) is

$$D = \frac{L^2}{900\,T}$$ where D = Total test deflection in inches.
L = Centres of spring in inches, measured along arc.
T = Thickness of thickest plate in ins.

or putting thickness of plate in sixteenth's = t

$$D = \frac{16\,L^2}{900\,t}$$

For unit deflection, $$d = \frac{0{\cdot}1\ L^3}{n\,b\,t^3}$$

where d = deflection in inches per ton.
b = breadth of plate in inches.
L and t as already stated.

Hence, to produce test deflection D, the corresponding test load

$$W_1 = \frac{16\,L^2}{900\,t} \times \frac{n\,b\,t^3}{0{\cdot}1\,L^3} = \frac{0{\cdot}177\ n\,b\,t^2}{L}$$

If W = working load on the spring,

$\frac{W_1}{W}$ = Factor of safety,

which should be 2·25 to 2·5, corresponding to a fibre stress in the plates of 30 to 26 tons per sq. inch.

These figures are for Carbon Steels to B.S. Specification 24/6. For springs made from Chrome-Vanadium or Chrome-Silicon alloy steels, about 25% more load is permissible. The unit deflection is the same for all practical purposes as for carbon steels.

In the formula for Test Load, we may insert the value, say, 2·5 for factor of safety, hence

$$\text{Working Load } W = \frac{Wt}{2{\cdot}5} = \frac{0{\cdot}177\ n\,b\,t^2}{2{\cdot}5\ L}$$

from which $\frac{n\,b\,t^2}{W\,L}$ = 14 (approx.), *i.e.* a spring giving a coefficient of 14 will have a fibre stress of 26 tons per sq. inch and a factor of safety of 2·5 (for carbon steel). An increase in the coefficient represents an increase in the factor of safety.

Springs are preferably made with plates all the same thickness. Plates vary from about $2\frac{1}{2}'' \times \frac{1}{4}''$ to $6'' \times \frac{5}{8}''$. The minimum number of plates is about 8 for locomotive springs. Two or three full-length plates are sufficient, and the length of the shortest plate may be (width of buckle + 2″) or L/n, whichever is the greater. Ribbed spring plates are now extensively used, thus avoiding the necessity for nibs and slots to locate the plates. Plates may be located in

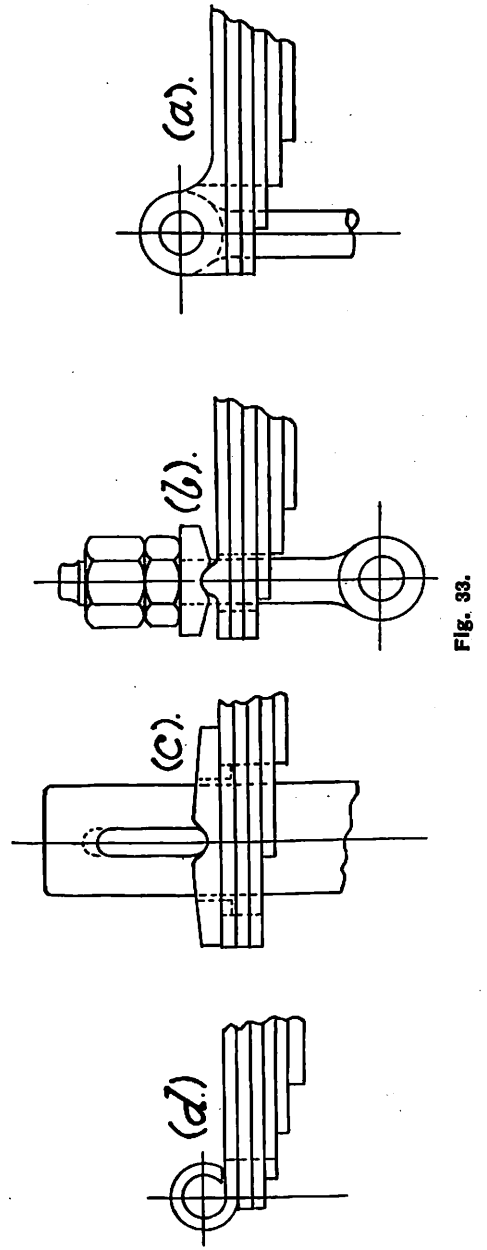

Fig. 33.

the buckle by a set screw and nibs in the plates, or by a rivet passed right through.

These centre nibs should be downwards, with the set screw in the top of the buckle. On underhung springs this may be difficult, but it is pointed out that upward nibs are a serious source of weakness, so that if downward nibs are not possible a rivet or some other fastening is preferable.

Buckles on underhung springs have a jaw in the top to take the tee-link. As the tee-link passes through the horn-clip, the design should aim at being able to remove the pin and get the spring off without having to remove the horn-clip or jack the locomotive up.

Overhead springs may sit direct on the axle-box or be carried on a pillar of some sort. If the springs are compensated, some arrangement to permit rocking longitudinally is required.

For attachment of spring links to the springs, four methods are common (see Fig. 33)—

(*a*) Top plate forged into an eye at the end, with a pin hole.
(*b*) Top plate jumped up into a small nib at the end, on which a bearing washer on the spring link rests.
(*c*) A bearing pad located on the end of the spring.
(*d*) Top plate rolled to form a pin hole.

Numerous variations on these types are to be found. Types (*b*) and (*c*) may be regarded as the best modern design.

The foregoing covers the main points of spring design so far as it concerns a locomotive draughtsman, as the actual detail design is largely in the hands of the spring makers. Where helical bearing springs are used it is desirable to refer their design entirely to the spring makers, as specialised knowledge is important with this type to get the best results. Readers who may wish to study the subject further are recommended to consult Mr. T. H. Sander's books on "Laminated Springs" and "Springs and Suspension," these being the modern standard works of reference on all matters relating to locomotive springs.

Spring Links.

Spring links are preferably made adjustable in length, so that they can be set to give the correct rail loads, the adjustment being made by nuts and bearing washers at one end. When links of fixed length are used, they are made with pins at the ends, or alternatively, made from flat bar with gib cotters taking the load.

The usual working tensile stress in a spring link is 2 to 2·5 tons per sq. inch, while the shear stress in the pins or cotters may be about 1·5 tons per sq. inch.

Spring Compensating Beams.

The position of the pivot on a compensating beam is determined from the loads on the springs which it connects. Thus (see Fig. 34) for equilibrium $W_1 \times x = W_2 (L - x)$, from which x is obtained.

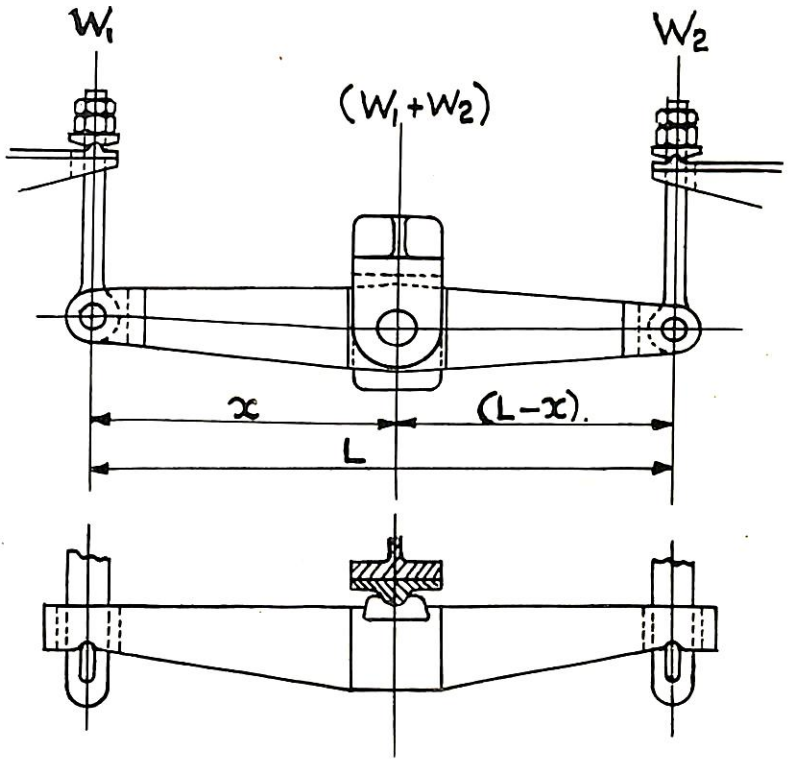

Fig. 34.

Or, taking moments about W_1

$$(W_1 + W_2) \times x = W_2 \times L \qquad \therefore \quad x = \frac{W_2 \times L}{(W_1 + W_2)}$$

For equal loads the pivot is central, *i.e.* x reduces to L/2 in the above formula.

The maximum bending stress in compensating beams is usually not more than $5\frac{1}{2}$ tons per sq. inch. The pivots may be pins, or knife-edge bearing pieces resting on hard steel pads. If pins are used, attention should be paid to their lubrication. The knife-edge bearing is best from a working point of view, as it has less friction than a pin. The ideal compensating gear should be sensitive enough to function rapidly whenever an increase in load occurs on one spring, so spreading the increase over the adjacent springs. If there is excessive friction in the rigging, combined with the natural inertia of heavy beams, etc., the action of the gear may become so sluggish as to defeat its object to a considerable extent.

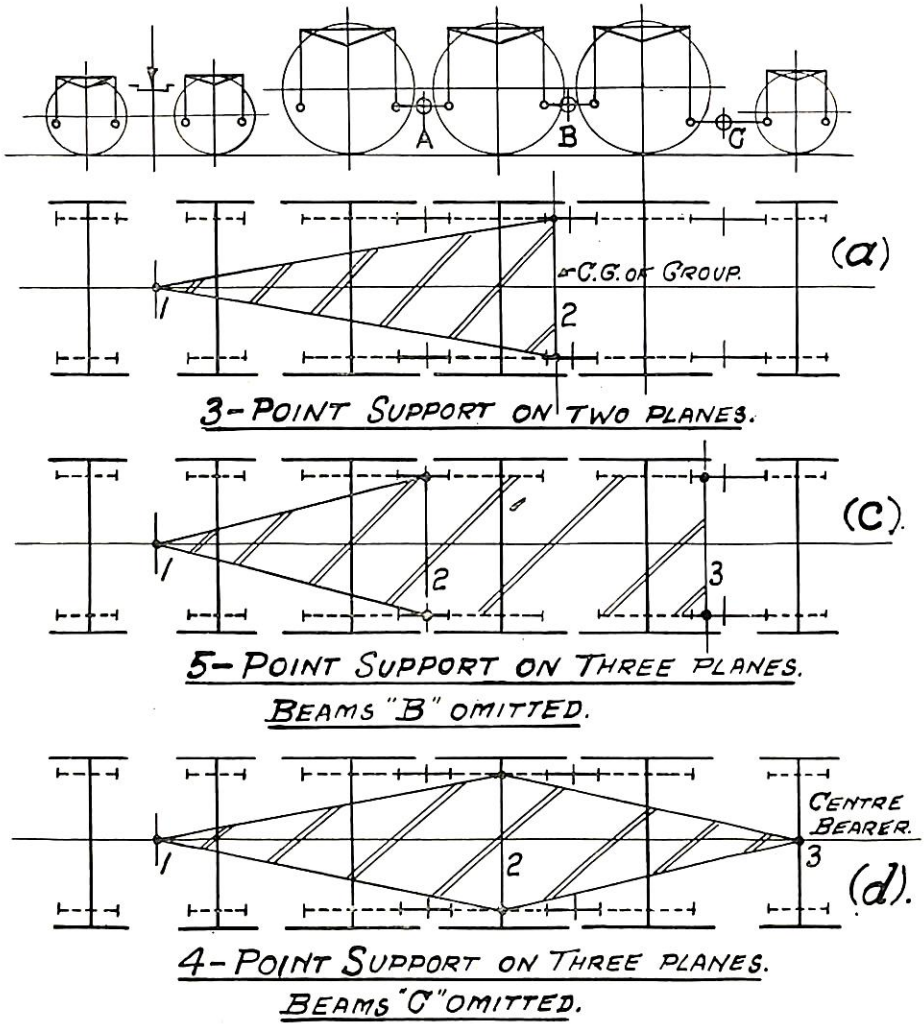
A
B
C
C.G. OF GROUP.
(a)
1
2
3-POINT SUPPORT ON TWO PLANES.
(c)
1
2
3
5-POINT SUPPORT ON THREE PLANES.
BEAMS "B" OMITTED.
CENTRE BEARER.
(d)
1
2
3
4-POINT SUPPORT ON THREE PLANES.
BEAMS "C" OMITTED.

Fig. 35.

Fig. 35 shows some diagrams of various arrangements of spring suspension for a 4-6-2 locomotive :

(*a*) 3-point suspension on two transverse planes.
Theoretically this is the ideal suspension, but on a very long locomotive the gear is heavy and sluggish in action and may even stick. Knife-edged bearings should be used to get the best results. This type of suspension is very extensively used in North America.

(*b*) 4-point support on two transverse planes.
As for (*a*), but the bogie supports the frame on two side-bearers.

(*c*) 5-point support on three tranverse planes.
Bogie is on a centre pivot as in (*a*).
Leading and Intermediate wheels in one group.
Trailing and Truck wheels in another group.

(*d*) 4-point support on three transverse planes.
Bogie on a centre pivot as in (*a*).
Leading, Intermediate and Trailing in one group.
Truck separate and on a centre bearing.

On British railways compensation is little used, except on the Great Western, where it is used to a limited extent. On Continental railways nearly all locomotives are partially compensated, generally on five points. The five-point suspension is probably best in practice, having shorter groups and fewer parts.

It would take up too much space to illustrate all the variations in spring arrangements for every wheel arrangement in common use. Drawings of all types can be found in the files of the *Railway Engineer*, etc., and in Mr. Sander's book on "Springs and Suspension," already mentioned.

Brake Gear.

Brakes on locomotive wheels are applied by steam, vacuum or compressed air. All coupled wheels are braked, and sometimes the leading bogie also. A four-wheeled leading bogie may have an independent cylinder or cylinders, while a two-wheeled leading truck generally has its brake gear operated from the coupled wheel group. On tender locomotives the tender frequently has vacuum brakes operating with the brake on the train while the engine has a steam brake. Alternatively, both engine and tender may have steam brake, with vacuum brake on the train. Where the engine is steam braked and the train vacuum braked, a special form of drivers' brake valve is frequently used incorporating a "proportional valve." When the driver applies the train brake by his brake valve (and also the steam brake on the engine) this device automatically regulates the power of steam braking in proportion to the power of the vacuum braking applied to the train.

Brake gears may be equalised or non-equalised. An equalised brake ensures a uniform load on all blocks, and requires less care when renewing blocks. On the other hand it contains many more pin joints. The majority of modern locomotives for main line work have equalised brake gear. For small shunting locomotives such provision is rather a luxury. Types of equalising gear are shown later.

The total pressure of the brake blocks on the wheels (always referred to as the Brake Power) is usually 75% of the rail load on the braked wheels with the locomotive in working order, except for tank locomotives or articulated locomotives, on which the rail loads are reduced as water and fuel are consumed. In the latter circumstances, up to 75% of the "mean weight" (with tanks and bunker half empty) is permissible. These figures represent what experience has shown to be suitable. It is worth noting that the coefficient of friction between the block and the tyre is not uniform, but increases rapidly as the vehicle comes to rest. This subject is receiving considerable attention in the present era of new high-speed trains, and higher brake power is being advocated to make it possible to stop the train from very high speed within a reasonable distance. However, it is mainly the brakes on the coaches which are being criticised, as it is safer to increase the brake power there than on the locomotive. On the locomotive there are variations of axle-load caused by rolling, bogies, unbalanced reciprocating weights, etc., which make it undesirable to increase the brake power very much. Some recent high-speed locomotives have been built in Germany with very high brake power, but they have brake blocks acting on both sides of the wheels. Such an arrangement has many advantages, but the rigging necessary to operate it is decidedly complicated when applied to coupled wheels.

The total leverage of brake rigging from the cylinder to the blocks should not exceed about $8\frac{1}{2}$ to 1 for steam and compressed air brakes, with ordinary types of cylinders having a stroke of 6″ to 8″. If a higher leverage is used the brake requires adjusting too frequently. This could be got over by using a special long stroke cylinder, but such an expedient involves a waste of steam or air when the piston is getting towards the end of its available stroke, and also a corresponding delay in the brake application.

For a vacuum brake, the difficulty of finding room for large cylinders sometimes leads to a higher leverage being adopted. The makers recommend the following :—

Cylr. Diam. × Stroke.	12″ × 6″	12″ × 8″	15″ × 6″	15″ × 8″	18″ × 6″	18″ × 8″	21″ × 9″	24″ × 9″
Max. Leverage	7	9	7	9	7	9	10	10

When calculating the brake power for a steam brake, it is usual to assume a pressure in the cylinder of 10 lbs. per sq. inch below the working pressure of the boiler, and the same figure may be used for the stress calculations. Alternatively, full boiler pressure may be taken for the stresses if one is very cautious, or if office records are made up on that basis.

For a Westinghouse Air Brake of standard type, the cylinder pressure is taken as 50 lbs. per sq. inch for the brake power, and 70 lbs. per sq. inch for stresses, as such an increase is possible in an emergency stop.

For a vacuum brake, a pressure of 10 lbs. per sq. inch on the nominal area of the cylinder is taken for both brake power and stresses. The actual pressure in a vacuum cylinder varies with the position on the stroke at which the brake comes on, and is reduced by repeated applications. Data on this subject is available from the makers, if required, but generally the figure of 10 lbs. per sq. inch is quite satisfactory for use on any normal design.

Brake Cylinders.

Vacuum and compressed air cylinders are made in standard types. The piston moves inwards in a vacuum cylinder, and it is carried vertically on a pair of trunnion brackets. (There is also a special double-acting horizontal vacuum cylinder now on the market). In a compressed air cylinder the piston moves outwards, and the cylinder may be placed at any desired angle. Steam cylinders are usually designed to suit the job. The piston may move outwards or inwards—preferably the former, as it avoids having to make a steam-tight gland. Some pistons are made steam-tight in the cylinder barrel by a flexible diaphragm, while others have the ordinary cast-iron packing rings. The latter is considered more reliable. A tell-tale hole near the end of the stroke gives a useful warning of need for adjustment.

Brake Gear Details.

The stresses allowed in brake gear details are approximately as follows :—

Pull rods—5 to 6 tons per sq. inch (direct stress).
Shaft levers, cross beams and brake hangers—8 to 9 tons per sq. inch (bending).
Shafts subject to combined bending and twisting—8 to 9 tons per sq. inch resultant stress.
Pins (double shear)—Maximum of $3\frac{1}{2}$ to 4 tons per sq. inch.
Pins (bearing pressure)—Not more than 9 tons per sq. inch, and usually much less.

The higher figures may be worked to when weight limitation or space restriction demand it. Pull rods less than $\frac{7}{8}$" diameter are

undesirable unless they are very short. Also, pins less than 1" diameter are not usually employed, as their wearing properties are poor.

Compression rods should be avoided, as they make the brake unstable in operation, with a tendency to bind on the jaws of the rods and so lose power through excessive friction. The exception to this is the brake cylinder piston rod, which is frequently in compression. For that reason it should be as short as possible. Pin holes and pins should be casehardened, and the section round the holes should include an allowance for wear and replacement by larger pins. Alternatively, hard steel bushes may be fitted in the holes, such bushes being not less than $\frac{3}{16}$" thick.

When designing a *brake shaft* the arms should be placed as near the bearings as possible, to reduce the bending moment on the shaft and save weight. When a shaft has arms at right-angles, a bending moment diagram may be drawn for the load in each plane and the resultant bending moment determined. Thus (see Fig. 36) if at some point on the shaft there are bending moments M_1 in one plane and M_2 in a plane at right angles, the resultant bending moment $M_R = \sqrt{M_1^2 + M_2^2}$. The point at which the maximum value of M_R occurs is usually obvious, especially from bending moment diagrams. Having found its maximum value, it is combined with the twisting moment T to find the equivalent twisting moment T_e.

Thus, by the usual formula,

$$T_e = M_R + \sqrt{M_R^2 + T^2}$$

If Z = modulus of section for twisting $= \frac{\pi d^3}{16}$ for a circular shaft, then $\frac{T_e}{Z}$ = resultant stress.

Note that although the twisting modulus of section is taken, the resultant stress is not a shear stress but an *Equivalent Tensile Stress* corresponding to the Equivalent Twisting Moment T_e in Rankine's formula above, of which the theory is that if the shaft breaks it will break by tension and not by shear. It has been shown that for a ductile material like mild steel the shaft is more likely to fail by shear. The theories on this point are fully discussed in Low's "Mechanics," and elsewhere, but it makes little difference in any case, as the stresses used in conjunction with the formula quoted here are known to be safe enough. This combined bending and twisting formula should not be applied indiscriminately, because if the bending is very small in comparison with the twisting (as when the shaft arms are right against the bearings), the equivalent tensile stress may look all right but the shearing stress due to twisting alone may be too high. It is desirable to check it both ways when in any doubt. The shear stress due to twisting only should not exceed about $7\frac{1}{2}$

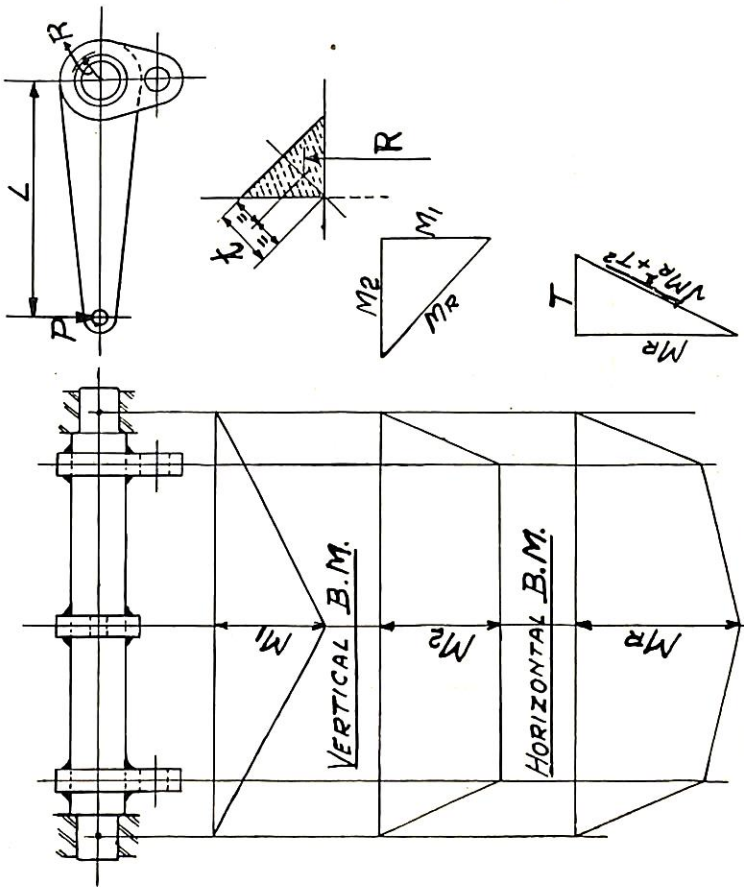

Fig. 36.

tons per sq. inch. Brake shafts with arms forged solid with the shaft are often specified, and require no comment. They are the most expensive type to manufacture. Alternatively, shafts may have loose arms fitted on squares, or loose arms bored out, shrunk on the shaft, and keyed. With the rapid development of electric welding technique it is now possible to build up a shaft entirely from bars, the arms being bored out to slip over the shaft, and welded in position with a heavy fillet weld on each side of the arm. The whole shaft is afterwards annealed, and the necessary machining done for bearings, etc. Such shafts are quite satisfactory in service, and effect a great saving in cost.

The welds joining the arms to the shaft must be of sufficient section to transmit the twisting moment from the lever to the shaft. Thus (see Fig. 36) if

P = load on lever,
L = length of lever,
R = mean radius of weld,
t = effective thickness of weld.

The shearing load on the welds $= \dfrac{P \times L}{R}$

Section of weld metal resisting shear $= 2\pi R \times t \times 2$
$= 4\pi R t$

$$\therefore \text{ Shear stress in weld} = \frac{P L}{R \times 4\pi R t} = \frac{P L}{4\pi R^2 t}$$

and the maximum allowable stress for this is about 3·5 tons per sq. inch for $\frac{3}{8}$" fillet welds and 2·5 tons per sq. inch for $\frac{1}{4}$" fillet welds.

Brake Hangers.

The space allowed between adjoining wheels has a great influence on the design of brake hangers, and a standard brake block of some sort usually has to be accommodated. (This matter should receive some consideration in the preliminary design of the locomotive when fixing the wheel-base).

Brake blocks are secured on hangers by various means, of which three are shown on Fig. 37—

(*a*) A pin through the block and hanger and a stop on the block below the pin to keep the top from rubbing on the wheel when the brake is off. This makes the block virtually a part of the hanger, and it must be made thicker at the bottom than at the top so that it can be worn right out.

(*b*) A "free" block, with a flat or helical spring to prevent the top from rubbing on the wheel.

(*c*) A "free" block, with a spring washer on the pin, binding the block to the hanger with sufficient frictional grip to hold it upright and prevent the top from rubbing on the wheel.

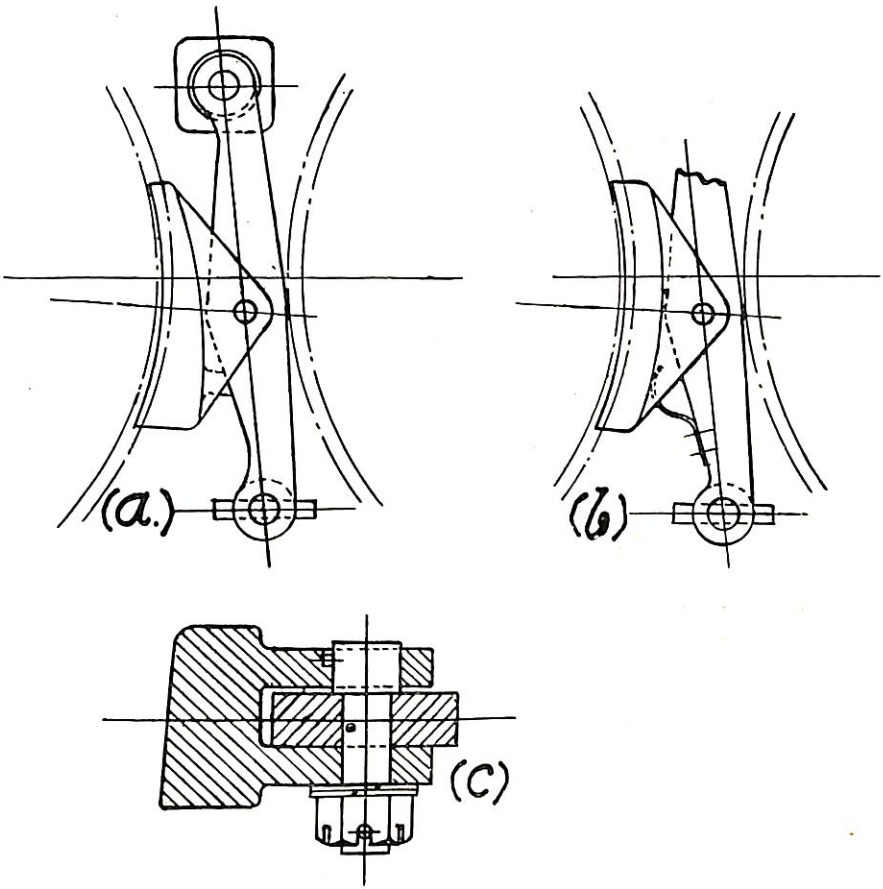

Fig. 37.

Type (*a*) requires blocks to be right and left hand, and the stops have to be chipped to suit when fitting the blocks, so that they will bear properly on the wheels.

For types (*b*) and (*c*) all blocks can be interchangeable (except for the drilling of type C).

When drawing a brake hanger and block, the following requirements should be allowed for in regard to clearances, etc.

The block to be at least $\frac{1}{8}$" clear of the tyre in the off position.

The relation of the block, hanger, crossbeam and wheel should be tried down with both the tyre and the block worn to scrapping size. This also determines the range of adjustment required in the main pull rods.

Wherever possible, the hangers should be designed so that the blocks can be renewed without uncoupling anything else, and the hangers should be inclined so that the block falls clear of the wheel by gravity when released.

The brake hanger bracket is overhung from the frame, and is subject to a bending load horizontally and a rather indeterminate vertical load from the frictional drag of the blocks on the wheel. This bracket should be generous in size, with a good base to the frame secured by three or four fitted bolts or cold turned rivets. Frequently the flange has a spigot formed on it which fits in a hole in the frame and reduces the load on the bolts or rivets.

Brake hangers should be bossed at each end to give a generous bearing surface on the hanger bracket and on the end of the cross-beam.

Bogie Brakes.

The wheels of a leading bogie or truck are sometimes braked. This may be done for the truck by a pull rod from the coupled wheel system, and if so it is essential that the brake gear should be equalised. It is possible to do the same for a four-wheeled leading bogie, but it is generally considered better to provide an independent gear. For steam or air brake this may take the form of a cylinder mounted on each frame operating directly on the brake hangers through short compression rods (see Fig. 38 a) the brake blocks being on the inner sides of the wheels. For vacuum brake (see Fig. 38 b) the cylinder or cylinders are placed between the bogie frames and are coupled to a cross-shaft, and thence to the hangers by compression rods (if the blocks are on the inside faces of the wheels). In its simplest arrangement this shaft is in fixed bearings and consequently the loads on the blocks are not equalised. However, the equalising can be carried out by introducing a floating lever and by proportioning the hangers to give the correct loads (see Fig. 38 c). Central compression rods may be used to transfer the load to the cross beams. This gives only two rods to be adjusted, instead of four as on some bogie brakes.

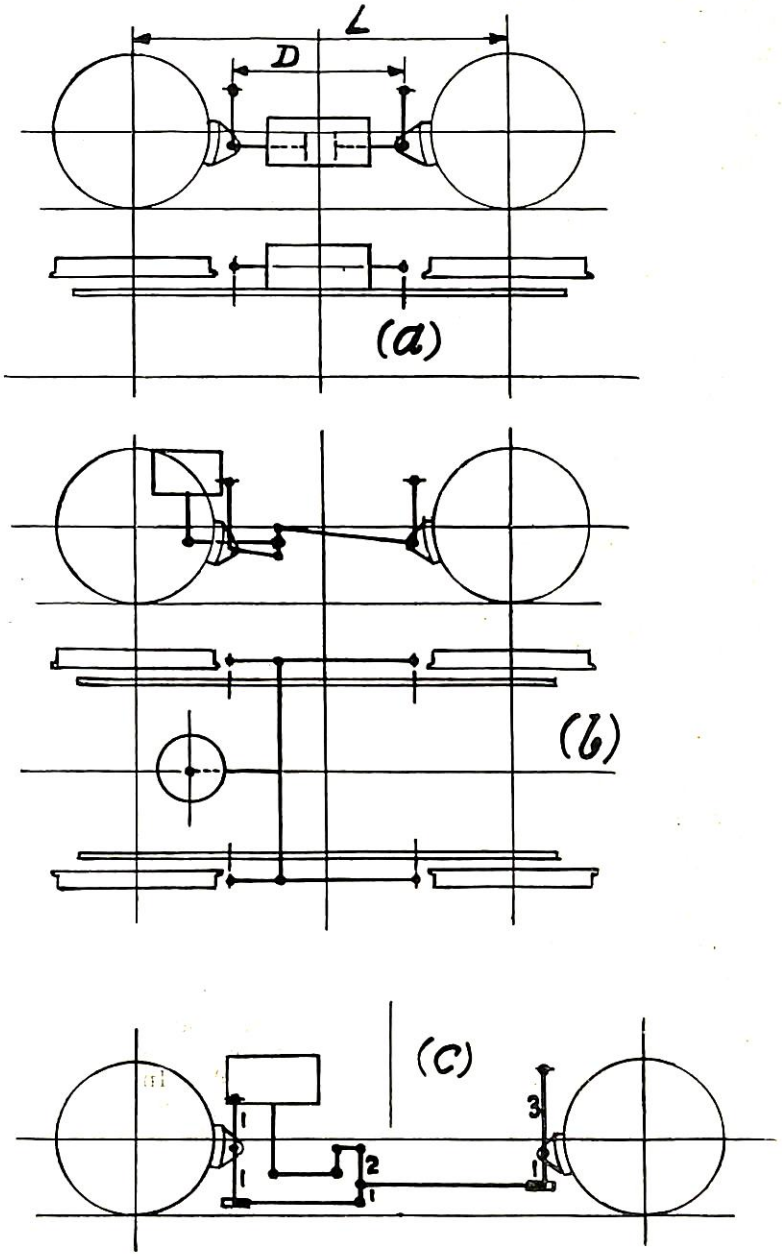

Fig. 38.

The compression rods are not desirable on principle, but they are the only simple arrangement for a bogie braked on the inside of each wheel. Stiffness is important in such rods, and a built-up tubular rod can be used with advantage and can be made to incorporate the screw-adjusting arrangement.

The location of the brake blocks on the wheels is important, because the frictional forces between the block and the tyre are transmitted to the bogie frames and affect the spring loads.

Thus (see Fig. 38 a) if

D = distance between brake hanger brackets on leading and trailing bogie wheels.
L = centres of bogie wheels.
W = vertical load on brake hanger.
= load on brake block $\times \mu$.

Couple applied to bogie frame = W $\times$ D.

This is balanced by an alteration in the loads on the leading and trailing bogie springs.

$$\text{Alteration in Spring Loads} = \pm \frac{W \times D}{L}$$

This alteration is directly proportional to the distance D, and hence the best arrangement of bogie brake is that with the blocks on the inner faces of the wheels, since this gives the smallest disturbing couple. If the blocks are applied to the outer faces of all wheels this couple is approximately doubled. It is not desirable to apply such a brake to a bogie whose springs are compensated, because of its tilting effect on the bogie frame.

Brake Equalisers.

Equalisers are usually interposed between the brake pull rods and the crossbeams. Two forms are commonly used—vertical and horizontal. (See Fig. 40). The leverage ratio of the equalisers is, of course, worked out according to the number of wheels to be braked, *e.g.*, for a group of eight wheels the first equaliser has a ration of 3 : 1, the second 2 : 1, the third 1 : 1.

With vertical equalisers, the best arrangement is to have a single line of pull rods on the centre line of the locomotive. Wear is taken up by adjustment at one point only—on the main pull rod. If the brake is a powerful one it is necessary to make the first equaliser at least, as shown, with the pin holes staggered and the rod ends fitted, one inside and one over the equaliser, to keep it down to a reasonable height. The vertical height should not be reduced too greatly by this means, however, because it is possible for equalisers to become tilted through variation in size of new blocks. This alters their effective leverage. The main pull rod should be about horizontal. If it is inclined upwards towards the crossbeam to any great extent the crossbeam is subjected to a vertical component of

the load being transmitted and will deflect considerably, as it is weak to resist vertical loading.

Instead of the centre pull arrangement, two lines of pull rods may be provided, giving two adjusting points. To make this arrangement completely equalised, two brake cylinders may be fitted, operating each set of pull rods independently through two loose levers mounted on a fixed shaft.

With horizontal equalisers two lines of pull rods are generally provided, and the whole rigging lies on the horizontal centre line of the crossbeams. The links from the equalisers to the crossbeams must be placed at the correct angle for equilibrium under load, as shown by the triangle of forces on Fig. 40. If they are not correctly placed, the rods will move laterally under load, and may foul adjoining parts of the locomotive. To make this arrangement completely equalised the independent cylinders and levers may be fitted as already described. An alternative arrangement sometimes used is to fit a cross equalising rod and bell cranks on the last crossbeam, as shown in Figs. 39 and 40, used in conjunction with a normal shaft having two fixed levers. Unequal adjustment on the two main pull rods is taken up by the cross-equaliser.

Adjustment of the main pull rods for wear may be made by a right and left screw-box ; by a screwed end on the main pull rod passing through a die-block in the brake shaft lever and secured by nuts ; or by a series of alternative holes in the end of the pull rod, staggered to give as fine an adjustment as possible. The latter type is not desirable for a powerful brake with heavy rods.

Fig. 39 shows details of vertical and horizontal equalisers, and Fig. 40 shows a diagrammatic layout for each type.

Frames.

Plate frames are used on all British railways, and on most locomotives of British construction. The thickness of plate used varies from about $\frac{7}{8}$″ to $1\frac{3}{8}$″ according to the size and weight of locomotive. It is impossible to say definitely what stresses act on a frame under running conditions, because of various indeterminate loads which may come on it. Hence the design of frames is largely a matter of experience and good judgment.

The depth of frame over the axle-box gaps should never be less than about 18″. The strength of the frame at this point and elsewhere may be checked under the following conditions.

(*a*) When wheeling the locomotive (with the horn-clips removed and the boiler empty), it should be possible to lift the locomotive by the buffer beams, or by special lifting shackles if provided, without the bending stress in the frames exceeding about 13 tons per sq. inch at any point. To determine this, an approximate bending moment diagram should be drawn, taking the approximate

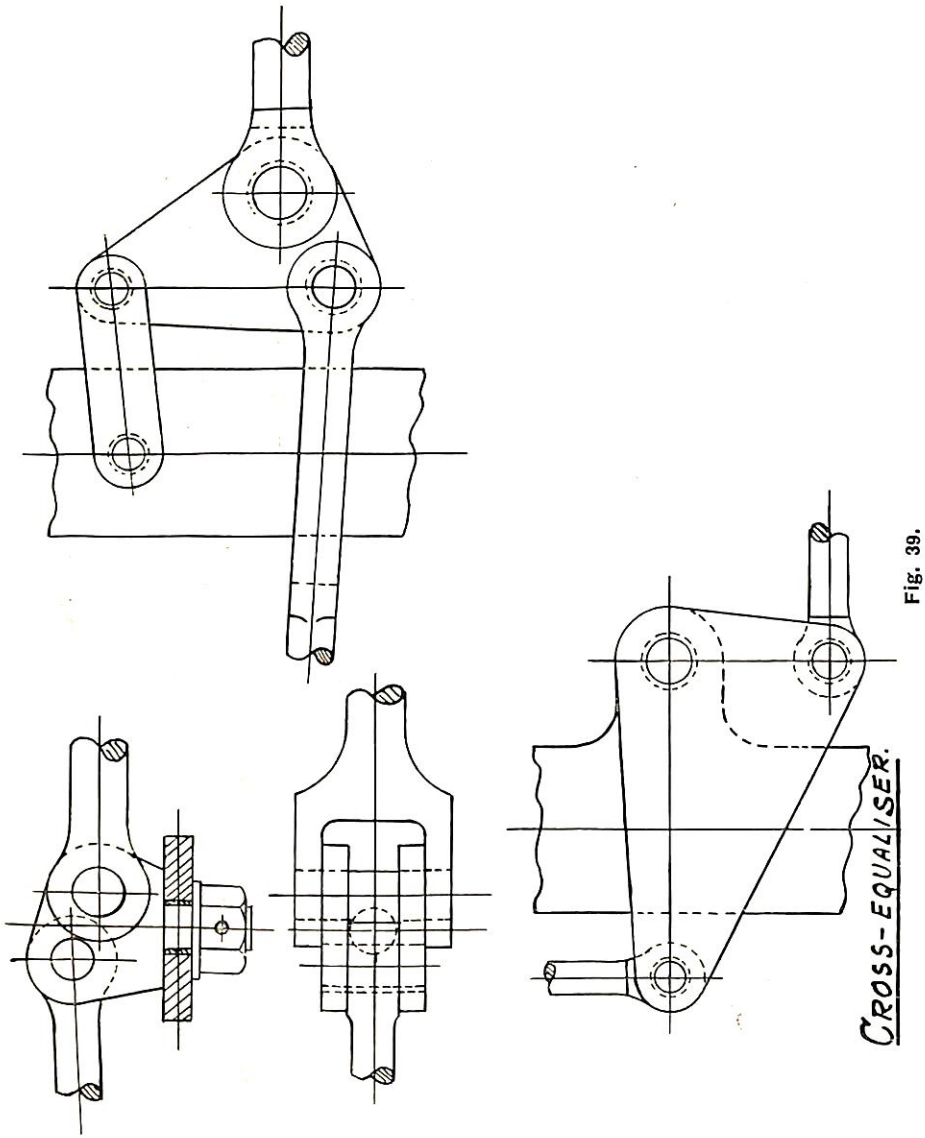

Fig. 39.

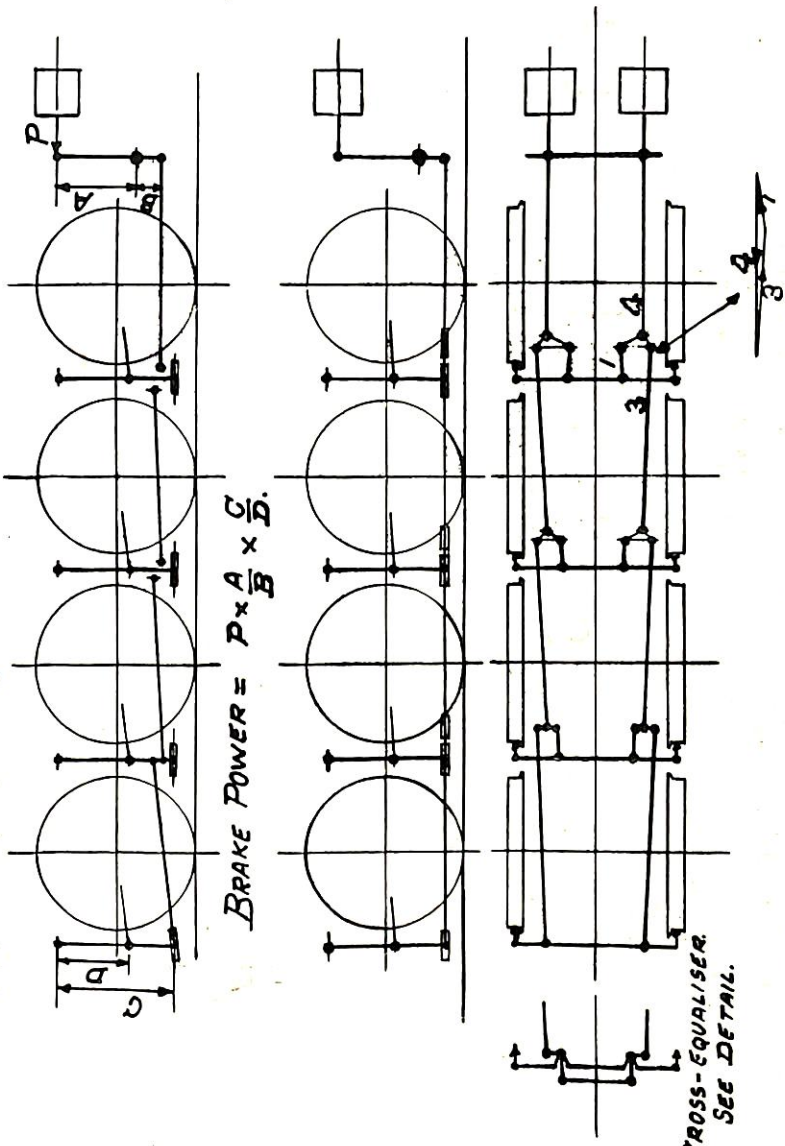
P
A
B
a
b
BRAKE POWER = $P \times \frac{A}{B} \times \frac{C}{D}$.
1
3
4
4
3
CROSS-EQUALISER.
SEE DETAIL.

Fig. 40.

weight of the boiler as carried at the centre of the smokebox saddle and the firebox supports, while the weight of the frames and attachments may be taken to be uniformly distributed.

(*b*) With the hornclips on and the locomotive in full working order, it must be possible to lift the locomotive, in the event of derailment, by means of jacks under the buffer beams (or at jacking brackets if provided). This puts the horn-clips in tension, and since their section is relatively small in comparison with that of the frame over the axle-boxes the maximum stress will occur in the horn-clips. It must be remembered, however, that in a very unsymmetrical beam such as this, having the weaker side in tension, the tension side will stretch much more than the other side will compress, with the result that the actual tensile stress will be less than that obtained by the usual calculation. In general, if the stress obtained by calculation is within the elastic limit of the material (say, about 13 tons per sq. inch), the section will be satisfactory.

Buffer beams are riveted to the frames with double angles and securely stayed with plates or castings to withstand the drawbar loads. Also, strong gusset stays are required outside the frame behind the buffers. The whole must be rigid enough to ensure that buffing and drawbar shocks will not distort the main frames.

Between the cylinders a rigid frame stay and saddle casting is fitted. (If the cylinders are between the frames they form a stay in themselves). Another substantial and rigid stay is required at the back end under the footplate. A vertical stay between the slide-bar brackets and a stay of some sort at the front of the firebox completes the list of stays essential in all frames. Further stays are fitted according to type of engine and conditions of service. Some railways specify a system of horizontal "racking plates," extending right back over the coupled axles, with attached vertical stays between each pair of wheels, this being the most rigid staying possible, and intended for rough service. On other railways a T-shaped stay between each pair of wheels is considered sufficient. One must be guided by existing practice, and apart from office drawings the files of the *Railway Engineer*, *Railway Gazette* and *The Locomotive* contain many useful illustrations.

Frame stays are riveted to frames with rivets $\frac{7}{8}$" or 1" diameter, in reamered holes. In the best practice, the rivets in the saddle-casting, slide-bar bracket stay, and some of the stays between coupled wheels are turned rivets, driven in and riveted cold. Where hot rivets are used, hydraulic riveting is desirable where possible, as this makes the rivet fill the hole better than pneumatic or hand riveting.

In recent practice steel castings for frame stays are to some extent being replaced by stays, fabricated from steel plates and bars by electric welding, whereby some weight can be saved. Heavy continuous welds are used, and quite complicated stays can be

built up in this way. The design must be studied from the point of view of welding, so that the welder may have proper access to all parts in their sequence of building up.

As an example of the advantage gained by using a fabricated stay, consider a simple flanged plate stay with angles riveted on to give a double row of rivets to the frame (see Fig. 41). The flanged plate is thinned at the corners in bending, and its flanges are thinned by machining, while the value of the angles depends on the strength and fit of the rivets fixing them to the stay. In the welded counterpart of this a broad flat bar forms the flange to the frame, the stay plate being welded to the flange by fillet welds on both sides. The thickness of the flange is independent of the stay plate, so it can be made to give any desired bearing surface for the rivets to the frame. Also, stiffening ribs, etc., can be added as desired.

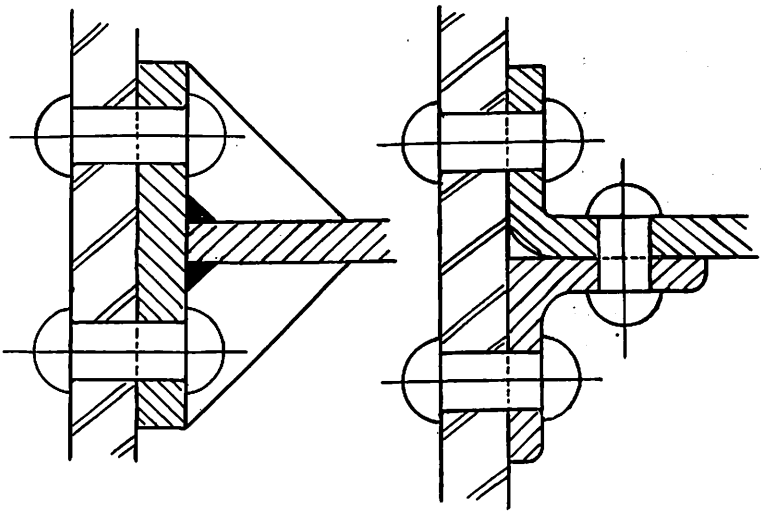

Fig. 41.

In general, it is desirable to arrange welds to be in shear only if possible, and not in tension. Holes in frame plates should be kept well back from the edges of the plate where possible, particularly on the lower edges. About 2″ from the edge for a $\frac{7}{8}$″ rivet is a desirable figure.

The fire-box normally rests on bearers on the top edge of the frame. On locomotives with a wide fire-box mounted over a trailing truck it is often necessary to provide a heavy cast steel stay at the front of the fire-box. This takes the weight of the fire-box on suitable bearers, and the main frames terminate at this casting. Side extensions on the casting give attachment for the remaining pieces of frame, which are set wide apart to suit the fire-box.

Bar Frames.

Apart from making bar frames of cast steel, as is done by some American companies, they are usually made from a solid steel slab, cut out and machined all over. The thickness varies from 3″ to 6″, and only a few stays at key points are required because of the natural rigidity of the frame itself. The usual practice for bar frame locomotives is to cast each cylinder integral with half the saddle casting, the whole when bolted together forming a combined frame stay and saddle casting.

Since the frames are weak in the vertical direction when the horn-clips are off, it is usual to tie them to the boiler barrel at points between the saddle casting and the fire-box. Also, at the front end stay bars are frequently fitted between the smoke-box and the end of the frame, to strengthen the frames for buffing shocks.

The frames are thinned at each end, the full thickness extending over the cylinders and all coupled wheels. Where a trailing bogie is fitted, a separate slab is sometimes used, let into a recess in the back of the main frame in front of the fire-box and securely bolted to it. The ties mentioned between the frame and the boiler barrel are made of thin plate (about $\frac{3}{8}$″) and are flexible enough to permit the boiler to expand longitudinally. A similar support is fitted at the back end of the fire-box.

Bar frames have the advantage of giving the greatest accessibility to all working parts between the frames. They require a minimum of stays, and the working loads from the piston thrust and the spring gear all act centrally on the frame, instead of being overhung loads as on a plate frame.

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